JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXIV. MAY, 1912. No. 2.

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THE FETISHISM OF SCIENTIFIC MANAGEMENT.

BY REAR ADMIRAL JOHN R. EDWARDS, U. S. N., MEMBER.*

INTRODUCTORY.

During the past two decades a considerable number of aggressive engineering experts have proclaimed the doctrine that with the adoption of scientific methods in the management of our railway and industrial corporations and establishments there would result increased profit to the employer, decreased cost to the producer, and easier instead of harder working conditions for the many employed in industrial pursuits.

It is a doctrine rich in promises, prolific in fancies and of such general attractive character as to appeal to the personal interests of the many. When analyzed, however, the system is found to be substantially an effort to extend the development

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of shop administration, cost accounting and business management by sequential progress methods and by a detailed, if not intricate, system of organization and operation.

The recognized leader of these efficiency experts is Mr. Fred W. Taylor, of Philadelphia, Pa., whose research work in the direction of developing advanced and progressive methods of tool making has made him one of the world-wide authorities upon that particular subject.

QUESTION DEVELOPING INTO ONE OF COMMANDING PUBLIC INTEREST.

Due to the personal efforts of Mr. Taylor and the engineering experts who believe in his methods, the subject of scientific management now commands public attention to such an extent that the question is being thoughtfully considered from sociological, industrial, political, financial and military standpoints.

The action of the United States Interstate Commerce Commission, in 1910, in investigating the request of the Eastern railroads to increase their freight tariff rates has had a very far-reaching indirect effect in causing extended and thoughtful consideration to be given to the study and scope of scientific management.

This investigation by the Commission extended over a period of six months. The railroads unequivocally maintained that higher freight rates were essential to meet their progressively increasing expenditures. In resisting this increase in freight rates various contentions were offered by the protesting commercial organizations. Many of the claims and contentions of the trade organizations were, in general, too technical to be understood by the public, but the catching epigram of Mr. Louis D. Brandeis, counsel of the Traffic Committee, that " at least a million dollars a day could be saved by the railroads by scientific management" caught the public fancy. From that day the question of scientific management of particular indus-

tries has been somewhat intimately associated with the more important problem of national efficiency. Railway officials, industrial managers and technical experts have been forced to meet the attacks and arguments of the efficiency promoters. Particularly within the past three years the scope and purpose of scientific management have been carefully analyzed, and as a result the subject is now receiving the extended and thoughtful consideration of many able engineering experts, industrial managers and Government officials.

BASIC PRINCIPLES OF THE SYSTEM.

Probably no clearer exposition of the system has been made than that presented by Mr. Henry G. Bradlee, a member of the well known Stone & Webster Company of Boston, a firm noted as expert authorities upon engineering and industrial matters. In a paper prepared for the Congress of Technology, Massachusetts Institute of Technology, Mr. Bradlee thus analyzes the basic principles of scientific shop management:

"Stripped of technicalities the method of the modern efficiency engineer is simply this—First, to analyze and study each piece of work before it is performed; second, to decide how it can be done with a minimum of wasted motion and energy; third, to instruct the workman so that he may do the work in the manner selected as most efficient.

"There is nothing fundamentally new in this method. The underlying principle is being used today to a greater or less extent in all industries and has, no doubt, been used at all times in the past. Let us keep this fact just as clear in our minds as possible.

"The method as employed by the modern efficiency engineer is distinctive, not because it is new, but because it is carried to much greater detail.

"The modern efficiency engineer is not content to plan out work along broad general lines. He proposes to go at it in a more scientific spirit. He plans to make a systematic study

of every detail and obtain maximum efficiency through preventing waste and loss at each and every point.

"With this in view he watches every motion of the workman's hands and body. If any unnecessary movement is made he tries to change the conditions under which the work is carried on or gives instructions to the workman so that the wasteful act may be avoided in the future. Every motion made and every bit of energy expended must be made to yield useful results in so far as this is possible.

"The form of organization adopted naturally has the same end in view. The number of overseers, supervisors, experts and specialists, in proportion to the number of workmen is materially increased.

"Special accounting systems are adopted to show at a glance what proportion of the cost of a piece of work is necessary and what proportion is caused by waste of energy. The information so obtained is used as a guide to prevent waste in the future.

"The workman is encouraged to coöperate through the use of a bonus system which aims to give the highest pay to the most efficient worker.

"These methods applied in certain cases have produced some very surprising and satisfactory results, but it is by no means a necessary conclusion that they can be universally applied with equal success.

"As I have already indicated, we are all of us familiar with the general principles underlying the methods of the efficiency engineer; many of us make frequent use of these principles in the conduct of our business. I think I am correct in saying that in the business with which I am connected every general principle and every detail method which has been suggested by the efficiency engineer has been used at one time or another and many are in use today.

"The subject is then a familiar one to all of us; the problem presented is not the adoption of something entirely new; but rather the extension to every detail of our work of something which we have already tried."

LITERATURE UPON THE SUBJECT.

The literature upon the subject is not very extensive, but is of striking character in two respects. The wholesale condemnation of administrative organization and shop management is its central key note. This spirit of condemnation is reflected in statements like the following:

"Inefficiency is not a local evil. . . . It extends through the whole of American life. . . . This inefficiency of effort pervades to a greater or less degree all American activities. . . . Inefficiency similar to that in the manufacturing shops exists in all building operations to the same or even greater extent. . . . The United States and State Agricultural Bureaus have determined like inefficiencies in farming operations. . . . In our whole educational system there is the same inefficiency."

There is no feature of American industrial affairs that appears to have escaped the condemnation of these efficiency managers. It is asserted that "The American railroad, by the most advanced engineering and industrial methods, carries an absurdly small net load for an absurdly small distance at an unnecessarily high cost." And yet in an address by Mr. Frank Trumbull, Chairman of the Board of Directors of the Chesapeake & Ohio Railway Company, he states:

"If you should write a letter to an American railroad official, his corporation will have to haul a ton of freight—2,000 pounds of average freight, coal, ore, silks, ostrich feathers and everything—for more than two and one-half miles to get money enough to buy a postage stamp to send you an answer."

With the wholesale condemnation of American efficiency, the literature is replete with superlatives telling of the efficiency accomplishments of the professional scientific management experts. These experts appear to have had the most remarkable success in directing the affairs of others, if their own statements and conclusions are founded upon facts.

The literature likewise abounds in platitudes, truisms and proverbs concerning the importance of efficiency and the necessity of preventing waste. The average industrial manager

has neither the desire, inclination nor time to controvert doctrines of such nature.

The trend of this literature does not appeal to intelligent labor nor does it impress industrial managers with its soundness of doctrine. Such expressions as "instruction cards for workmen," "time studies," "differential piece-work system," "stop-watch studies," "speed bosses," "tasks" and "systematic soldiering" do not commend themselves to the self-respecting mechanic. The wholesale denunciation of existing industrial management is resented by administrative officials.

The promises of higher dividends, reduced operating expenses and augmented output naturally appeals to every stockholder who is looking for increased returns upon his investment. As in the case of other emotional propositions, however, the system is more successful in obtaining converts than in holding them to the faith.

One is surprised, if not astonished, to find that the supporters of scientific management have been permitted practically to take unto themselves the literary field in regard to industrial management. It has only been within the past two or three years that there has been even any individual earnest effort to controvert the doctrine. The neglect to state the other side of the proposition has probably been due to three reasons: First, the general inclination to encourage and favor any effort to promote efficiency and reduce waste; Second, the indifference of many manufacturers and managers to the work and purpose of the efficiency experts, it being considered that the agitation was of an ephemeral nature and that it was not founded upon either fact or experience; Third, the contempt with which many executive officials resent the general proposition that the average scientific management expert could secure help, go to a mill or enter a shop and solve problems which those in charge of management had not been able to solve with years of experience, with patient research work and experiment, and with the benefit of traditions of the business.

Conservative industrial leaders are just coming to a realization concerning the depth and extent to which the doctrine has

soaked into the public mind and the marked favor in which the system is held by certain elements of the general public. It is quite probable that from henceforth the literature upon the subject will not be of such one-sided character.

THE SCOPE OF THE SYSTEM.

The efficiency experts primarily intended to confine their operations to effecting improvements in shop management, but during the past few years their aspiration and scope have greatly broadened. The most progressive and competent of them now maintain that scientific management is more necessary in the conduct of administrative affairs than in that of shop management.

It is significant that the trend of development is thus in the direction of securing a more efficient administrative staff. The effort to commence reforming the methods of the unskilled laborer seems to be giving way to the purpose of securing a more systematic and efficient industrial directive organization. It is therefore probable that the business researcher will supersede the efficiency expert, since success in promoting efficiency is more likely to be obtained by a comprehensive study of every feature of an industrial organization than by assuming that the essential weakness of management can be located in the producing department.

Concentration of effort in the study and investigation of manufacturing and operating problems alone will never meet the demands of successful business administration. The economies strained after in the production department are often of minute character. In reference to this matter, Mr. H. F. Stimpson, Chief Engineer of the Universal Audit Company, New York City, thus writes:

"Practical examples of the mistake of putting operating experience in the saddle will come readily to mind. In steel manufacture they are frequently measured in cents per ton of product. Yet mistakes in cost finding, in the price at which product is sold, in the price paid for materials, or in credit

improperly extended, may often neutralize all that the best equipped manufacturing department has accomplished in lowering the unit of production cost. President Farrell, of the United States Steel Corporation, said in an address before the New York meeting of the American Iron and Steel Institute:

"'Without any desire to belittle the practical value and importance of effecting a saving in cost production, however small, it has always seemed particularly hard on the men who have accomplished what may appear as an insignificant reduction of the previous month's costs of producing pig iron or ingots or wire rods, to have this saving in mill costs dissipated by the sales department 'quietly meeting the market' with a reduction of $1 or $2 per ton at the first sign of any cloud on the commercial horizon. Good salesmanship and sound business principles in the conduct of the selling of iron and steel are just as essential and vital to the low cost as up-to-date machinery and manufacturing practice.'"

BASIC PRINCIPLES OF SYSTEM ENUNCIATED OVER A CENTURY AGO.

In reference to the development of the system Mr. John Calder, Manager, Remington Typewriter Works, thus comments:

"The minute sub-division or processes in manufactures was predicted and its advantages set forth in 1776 by Adam Smith, the Scotch philosopher at the University of Glasgow and the gifted author of 'The Wealth of Nations.'

"Sixty years later the principle was firmly established, and Charles Babbage, the noted English mathematician and mechanician, described in 1834 in his 'Economy of Machines and Manufactures' the minute division of labor possible in repetition work in his day in various industries. He also furnished a complete philosophy of the subject and examples of calculations as to the limits of reasonable investment in labor-saving machinery.

"Industries, such as textiles, in which machinery reigned supreme at a very early stage, were most affected by the new principle which evolved quite naturally with the dawn of modern industrialism.

"In our own day practical political economy has been somewhat neglected by engineers, and three-quarters of a century after Babbage we find the division of labor by machines carried much farther than the divisions of handicrafts which he also advocated and described."

LIMITATION IN EFFICIENCY METHODS OF OPERATION.

Even where conditions appear favorable to efficiency methods we find important limitations to their adoption. In regard to this special feature of the question, Mr. Bradlee, of the Stone & Webster Company, thus comments:

"Low cost of operation or of manufacture is, after all, only one factor out of many to be considered in measuring industrial efficiency. It frequently happens that the lowest cost can only be secured through sacrificing other and more important factors. Let us consider some examples.

"We use a special delivery stamp, send a telegram in place of a letter, or ship merchandise by express instead of freight because saving in time is more important than saving in expense, or because there are advantages in extending our business over a considerable area and this can only be done by using these methods.

"The steam railroads increase their operating costs per ton mile by operating express service. By doing this they have helped to build up industries which could not otherwise exist. We are glad to pay this extra cost so that we may no longer be dependent on a local supply of fruits and other perishable goods.

"In construction work we frequently adopt methods which might be considered extravagant if we overlooked the advantages which come from the completion of the work by a

certain date. Delay in completion is often far more serious than quite a considerable increase in cost of work.

"It is often more economical for a street railway to attach trailers to its regular car to handle rush-hour business than to operate additional motor cars. The public, unfortunately, do not like trailers and, here again, the railway decides that public good-will is more important than a slight saving in expense.

"A very simple case will illustrate how efficiency in one direction may conflict with efficiency in another. The crew on a locomotive have three duties: first, safety of the train and its contents; second, the maintenance of schedules, and, third, operation of the locomotive at the lowest possible cost. Let us suppose the railroad is making a special effort to improve fuel economy. The locomotive crew became very much interested in the matter and the first year they succeeded in saving several hundred dollars worth of coal. The second year they decide to do even better, but one day when they are trying to make a particularly good coal record they run by a signal, wreck the train and kill a dozen passengers. How shall we measure efficiency in this case? Coal efficiency is high, accident efficiency is low; the two are always somewhat in conflict. It would have been much better for this road to have burned a little more coal and avoided the accident.

"The spirit which runs through an organization, its esprit de corps, is an important factor in its success or failure. A superintendent or foreman who has the faculty of keeping his men always happy and contented, even though he is, at times, somewhat extravagant, may be more valuable and more truly efficient than another who is able to get a little more work out of his men but who keeps them continually growling and grumbling against the business and their employer.

"In these few examples we see that diversified industries, public health, safety and welfare, speed of action, time of completion, esprit de corps, quality and quantity of service, public good-will and patronage; all these and many others enter into measurement of success and efficiency."

SYSTEM ENTAILS EXCESSIVE NUMBER OF RECORDS.

Ex-Senator Aldrich, who in many respects was the ablest business man that ever entered public life, declared that it was through simplification of government business methods that the most material reduction in national expenditures could be brought about. The several executive departments of the Government, as well as the leading railway and industrial corporations, are progressively reducing the number of their records.

The trend of scientific management, however, is to increase the clerical force several fold. This feature alone has a tendency to discredit its value in the minds of those who believe that too many, rather than too few, records are now being kept.

Several years ago one of the most noted engineering experts of the country was retained to revise shop methods and improve the product of an established concern whose business had been progressively decreasing. It appears that it had been the practice of the firm to attach exceptional importance to the work of determining in detail the cost of each part of every article mannfactured. The engineering expert found so many records and card indexes that he epigrammatically stated that it appeared to him as if the man who had designed such a cost accounting system must have had a personal interest in some stationery establishment.

There is a tale extant of a certain flag lieutenant in the Navy who, upon being told of the fruitless night-and-day effort of his predecessor to dispose of matters pending, threw all these records into the sea with the remark that he would start with a new slate. The story goes that there were but few of these documents concerning which further inquiry was ever made; and that when special inquiry was made regarding a particular paper, the officer making inquiry was simply directed to start the correspondence anew.

The trend of business management has been to simplify the

character and reduce the number of records, and surely scientific management is not in line with such practice.

SUPERVISION AN IMPORTANT FACTOR IN INDUSTRIAL MANAGEMENT.

An important element of industrial success is efficient, systematic and complete supervision by the administrative and technical officials of the plant. Where intelligent, considerate and capable supervision exists the best energies of the employés are aroused. Scientific management tends rather to minimize supervision by substituting a very comprehensive system of planning and routing the work. Experience shows that the most efficient method of keeping track of the work is by personal supervision, and not through a system of cards and records.

One of the most successful features of the shop management of the Baldwin Locomotive Works is the exceptionally efficient and capable supervisory force employed. In this establishment it is not considered that the resourcefulness and inventive talent of the plant is concentrated in either the planning or drafting rooms. Simple common-sense methods are employed in routing the work, and the foremen and quartermen are not burdened with the work of looking out for a mass of records as regards the time and cost of making detailed parts. More accurate information is obtained from mass costing.

The average workman wants to do right, and he is more content to have his work judged by competent and conscientious supervisors than by intricate records compiled and tabulated by clerks in the planning or auditing offices who can have no conception of his technical skill, experience and resourcefulness.

Where extended personal supervision is conducted, there exists definite information concerning the character, skill and efficiency of each individual employé, and thus the idlers can be selected for discharge when work is slack.

PLANNING AND ROUTING OF WORK.

In analyzing much of the literature published upon the subject, one notes the exceptional importance attached by the advocates of the system to the planning and routing of the work through the shops. In fact such a degree of importance is attached to this phase of shop methods that the actual production of the article or accomplishment of the task appears as a small factor compared with the manner in which the work is done. Planning and routing work in a simple, practical and systematic manner has been coincident with shop development and extension, and the effort to attach undue importance to the planning and routing of work will not stand the test of time.

Upon this phase of shop management, London "Engineering" thus comments:

"Scientific management postulates a system in which every movement of every workman is the subject of written instruction supplied to him on cards, in which every tool to be used and the speed of every machine is regulated by a special department, which again deals with the workmen who are using the tools and running the machines, by means of written instructions. It postulates a system in which every workman is subject to instructions from five different foremen and three distinct sets of clerks, each dealing with one part of his work only and each acting independently from the others; and it postulates a system in which the route of every piece of work through the shops is the subject of daily written instructions from a special department, and in which bodies of men and gangs are similarly controlled in their movements by a special department, which in some cases works out its instructions by means of a sort of diagrammatic chess-board arrangement.

"We have no quarrel with any of the methods on which scientific management is based if they are used within reason, but we simply do not believe that such a system as is outlined above is a practicable one for every-day life. It introduces some seven new links into a chain which under ordinary meth-

ods contains about four, and with these seven new links must introduce seven new possibilities of efficiency and failure. At each of the new stages there is always a chance of small delays and losses creeping in, and with ordinary men under ordinary conditions these delays and losses will creep in. The whole organization is of such complexity and delicacy that a little derangement of one of its parts will upset the whole working, while its expense must be such that savings over ordinary methods can only be shown if it works consistently at its best. The setting in motion of such a system of management shows a lack of appreciation of the capabilities of ordinary men for whom systems are made, and, in our opinion, any attempt at its general adoption can only end in disappointment."

CONFIDENCE AND COÖPERATION.

During the recent great railway and coal strikes in Great Britain, one of the most noteworthy features observed by visitors was the unswerving confidence of the people in the ability of the Government to handle the situation. The reason was simple. The railway administrators, the various labor unions and the general public had implicit confidence in the integrity, justice and common sense of the commissions appointed to arbitrate the various matters.

In some respects the most objectionable feature to the introduction of the system of scientific management is the lack of confidence and respect which it breeds between capital and labor. It is hammered into the employer that soldiering and loafing is the general rule upon the part of the workmen, while the employé is told that inefficient and wasteful methods of management prevail everywhere. The first impression that the ordinary manager would form upon reading one of the standard books is the general lack of efficiency, loyalty and zeal upon the part of those in his employ. In turn, a cursory reading of the same book would probably cause the average workman to believe that the organization and shop management of

his employer are not in accord with modern business methods. Such literature is not conducive to mutual confidence between employer and employé.

INDUSTRIAL OPERATION AND ADVANCE DEPENDS UPON MANY FACTORS.

In the conduct of every enterprise it is essential that rules and regulations, based upon business experience and sound judgment, should be established and regularly maintained. There are certain essential features of business management that must be considered in the organization of every shop, whether large or small, among which are the following:

Staff and departmental organization.

Care of stores, stock, tools and machines.

Systematic methods of keeping time.

Regulations concerning discipline and methods of work.

Wage and premium system; rates for overtime.

Use of bulletin boards and suggestion boxes.

Systematic method of noting the progress and cost of work.

It is unnecessary to state that the detailed list of items of essential elements of shop management could be extended indefinitely. The items enumerated above are simply mentioned as illustrative of the fact that the ordinary shop should be, and probably is, operated under fairly good rules. The scientific management experts are, therefore, no more responsible for the introduction of ordinary common-sense system of cost accounting and efficient methods of shop management than they are responsible for the installation in modern shops of either the telephone or the electric light.

Where a remunerative business is being carried on and the establishment has no spirited competitors, it is exceedingly probable that certain lax methods may prevail both as regards shop practice and management, and any practical expert could probably offer suggestions that would promote efficiency.

Increasing the ordinary capacity of a plant which has to seek

trade against keen and aggressive competitors is a problem that is more likely to be solved by those familiar with the industry than by one unacquainted with the business. It requires an expert of extended experience to effect improvements in detail, and it is not surprising, therefore, that nearly all connected with industrial development refuse to accept the doctrine, without protest, that improvement can so easily be brought about by men unfamiliar with the work.

As regards improvement in cotton mills, Mr. Henry D. Martin, Superintendent of the Lancaster, Mass., Weaving Mills, declares that the earning capacity of a plant depends upon many distinct features such as,—

1. Convenient location of plant.
2. Proper layout of buildings.
3. Right kind of machinery.
4. Correct arrangement of machinery.
5. Uniform engagement of the processes.
6. Purchase of good raw material at lowest price available.
7. Skilled manufacturing organization.
8. Homogeneous service of the entire force.
9. Maximum production.
10. Minimum account of seconds.
11. Minimum waste account.
12. Lowest degree of wear and tear.
13. Sufficient expenditure to keep machinery in good running order.
14. Lowest cost of production.
15. There must be a good market for the finished product at fair selling price.

It might not require extended experience or marked ability in management upon the part of any expert to note wherein improvements might be effected as regards certain features of administration. The introduction of such changes, however, might interfere with the operation or impair the effi-

ciency of other features of the plant. Criticism and condemnation as regards special features are therefore unwarranted and unjustifiable unless the whole fabric of organization and management is considered in relation to apparent defects in the management of special shops of the industry.

The effort to increase the producing capacity of every man and machine connected with manufacturing establishments has been coincident with the development and extension of every industry. The suggestion that the country possesses a contingent of efficiency experts who, after a few months study of an industry, can effect economies and improve the character of the article manufactured has met with but little approval in the industrial world. It may be possible for such experts to familiarize themselves with certain important features of an industrial plant, but the lack of special knowledge of even a few of the elements entering into the earning capacity of such industries would undoubtedly baffle any attempt of the average efficiency expert to effect any permanent improvement, or to bring about any substantial economies.

REPETITION VERSUS REPAIR WORK.

Probably no more effective argument could be advanced against attempting to apply an extensive system of scientific management for varied or repair work than by briefly showing the general character of shop methods requisite for repetion work.

The Remington Typewriter Works is one of numerous concerns manufacturing thoroughly standardized products which, for periods of at least a year, they will not modify. In the conduct of such a business the special difficulties experienced by the ordinary industrial establishments are not encountered. Where repetition work is carried on, therefore, close attention can be concentrated on a limited number of definite problems, the satisfactory solution of which may be attained by gradual and experimental stages.

The Remington typewriter contains about 2,500 highly-

finished individual pieces, each of which passes through from six to thirty manufacturing operations. A total of about 1,000,000 interchangeable metal pieces, of several thousand varieties, are produced daily, and built each day into one of the various standard appliances turned out by the Remington Company.

The buildings connected with the plant contain about 1,600 machine tools operated by mechanical power. About 60 tools are operated by hand labor. The equipment likewise contains a large quantity of finely made and costly jigs, tools, fixtures and other labor-saving devices. Many of the machines are of such special design that they have had to be constructed at the works. While the greater portion of the machine tools are of standard make, it has been found essential to fit a considerable number of them with special attachments whereby repetition work can be carried on in an intensive manner. It is unnecessary to state that such an equipment of machinery is not found in any establishment where work of varied or repair character is carried on. The manager of the works states that every arrangement of shop methods has been based on business facts and careful analysis, and not on any preconceived fancies. No scientific claim is advanced for the management, since the system is modified in its details whenever circumstances warrant such change.

About 1,700 men are employed in the manufacture of the various parts of the machine. About 500 additional men are required to assemble the machines and to repair tools. These shops run almost automatically since seven-tenths of the employés are assured of steady employment. It requires but 20 persons to look out for the accounting as well as for the labor and motion-study sections. It will be observed that this small contingent of employés not only plan and route the work but likewise look out for the accounting. As regards planning, routing and cost accounting, the manager of the works, in an address before a joint meeting of the Engineering and Railroad Clubs of Altoona, thus spoke:

"*Planning.*—I do not favor the running of shops such as those here described, or indeed any class of manufactures, by fiat in every detail from a central planning department. Attractive as a general scheme, the complete centralization of the initiative and thinking of a plant is neither 'scientific,' economical nor practical. If attempted, to square with some theory, it undoubtedly slows up the plant and lessens the internal 'good-will,' a most valuable asset.

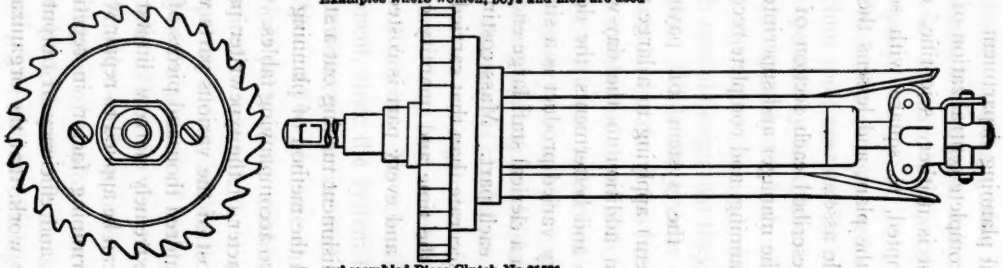
"In the works system here described each section of the organization in conference with the manager and superintendents has its appropriate part in planning, and complete records of the same are readily accessible.

"*Costing.*—It follows from the system of payment (straight-forward piece-rate system) applying to a large proportion of the employés that, in addition to the day-work accounting on plant maintenance and betterments, the determining of the direct cost of a very varied product is a simple matter. It is needless to maintain a clerical staff large enough to cost every recurring order for each part. Mass costing is found on this system to be more accurate than that carried out continuously in great detail, but the time and quantity cards are accumulated for every order, and every part is costed by selection at intervals."

The manner in which an establishment turning out a standardized product is organized, and the method of planning and routing work are shown on the two accompanying tables. One has but to note the size and character of the typewriter parts, together with the fact that the cost of the various parts were obtained from the record of a hundred thousand pieces of the same dimensions, to show the extremely narrow limitations within which repetition methods can be applied to repair work.

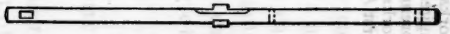
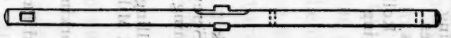
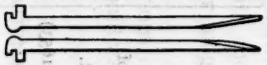
It is because cost is the determining factor in repetition work, while speed of completion and efficiency of output are the determining factors in repair work, that the organization and methods of the one plant cannot be duplicated in any essential features by the organization of the other.

DIVISION OF LABOR AND SPEED OF WORKERS IN ASSEMBLING SMALL INTERCHANGEABLE PARTS
Examples where women, boys and men are used



Assembled Piece Clutch No.31531

Stage No.	Unit Parts 18 in Number Assembled in 10 Stages	Operations in Sequence	Grade of Labor	Average Time Including Stops, from Records of 100,000 Pieces	
				Min.	Sec.
1		Ream 2 holes 0.073" in Roll Holder No.30670 on speed lathe	Women	0	7
2		Assemble roll holder No.30670 2 rolls No.30680 and 2 pins No.67640 at bench	Boys	0	45
3		Ream 1 hole 0.166" in clutch frame No.30361 on speed lathe	Women	0	5

4		Tap 2 holes 0.06″-56 in clutch frame No.30361 on tapping machine	Men	0	16
5		Ream 2 holes 0.085″ and 0.059″ in shaft No.31620 on speed lathe	Women	0	10
6		Straighten shaft No.31620 on bench block	Boys	0	8
7		Put pin No.65610 in shaft No.31620 with bench fixture	Boys	0	20
8		Put in washer No.90000 and assemble tube No.31521 and spring No.67210 at bench	Men	2	24
9		Bend and gage clutch jaws No.30690 in bench vise	Men	0	31
10	See full-size sketch	Assemble clutch jaw No.30690 ratchet No.30370, frame cover No.30330, 2 screws No.70402, and roll holder comp., with pins No.64650 at bench	Men	3	53
	The direct labor cost of assembling 1 clutch No.31531 is 3.784 cents	Average time to assemble 18 unit parts to form clutch No.31531	Women Boys and Men	8	40

ORGANIZATION OF REMINGTON TYPEWRITER WORKS.

The organization is indicated on the chart by the numbered and grouped functions which the various assistants control. The figures, in parentheses, below each division on the chart indicate the total number of employés engaged in that section.

1.—*Manager* (2500)

2.—*Executive Departments* (363)

- 4. Customers' Orders and Shipping (42)
  - Correspondence on Orders
  - Stock of Finished Machines, Packing Room
  - Shipping and Billing Machines and Parts
- 5. Purchasing and Stores (34)
  - Purchasing Raw Material and Supplies
  - Receiving Clerk, Teams, R. R. Siding
  - Raw Material and Supply Stores
- 6. Finished Parts and Production Orders (18)
  - Finished parts Stock Room, Simple and Compound
  - Issues all Production Orders for Parts and Mat.
  - Issues all Parts for Machine Manufacture
- 7. Works Engineer'g (90)
  - Light, Heat, Power, Accident Prevention
  - Millwrights, Electricians, Plumbers
  - Fire Corps, Watchmen, Yard Labor
- 8. Designing and Tool Engineering (106)
  - Inventing, Model Making, Specifications, Catalogue
  - Designing Parts and Jigs, Toolmaking
- 9. Inspection (35)
  - Materials, Work in Process and Finished, New Tools
- 10. Accounting, Labor and Motion Study (20)
  - Cashier and Bookkeeping
  - Costing and Inventories
  - Industrial Betterments
  - Time Records and Pay Roll
  - Rate and Method Testing
  - Works "Bulletin"
  - Suggestion System
  - Foremen's Club House
  - Works Band
  - First Aid

3.—*Manufacturing Departments* Superintendent (2137)

- 11. Manufacture of Parts Assistant Supt. and Foremen (1653)
  - Pattern Shop
  - Brass Foundry
  - Iron Foundry
  - Japanning
  - Tinsmith
  - Drop Forging
  - Punch and Press Work
  - Automatic and Hand Screw
  - Annealing and Hardening
  - General Grinding
  - Light Drilling and Milling
  - Cast Iron Drilling and Milling
  - Polishing
  - Plating and Buffing
  - Wood and Rubber Working
  - Typemaking
  - Assembling Compound Parts
  - Tool Maintenance
- 12. Manufacture of Machines Assistant Supt. and Foremen (484)
  - Assembling Machine Frame Group
  - Assembling Machine Type Bar Group
  - Assembling Machine Carriage Group
  - Assembling, Final Stage
  - Adding Special Features
  - Rebuilding Machines
  - School of Repairing

MEN ARE MORE IMPORTANT THAN MACHINES.

One of the fundamental weaknesses of scientific management is due to the importance attached to the efficiency of the machine as compared with the efficiency of the man. The man frequently appears to be regarded as a hopper for feeding or watching the machine, it being considered that the shop foremen and planning experts will supply all the brain work required.

The fact seems to be overlooked that the distinctive features of modern industrial advance have been the resourcefulness and observing talent of the individual. Any attempt to dispense with the experience and originality of the artisan is marked retrogression. The shop practice of the past generation invariably placed the man above the machine, and therefore the best energies of the most successful administrative officials have been in the direction of arousing the interest and latent resources of the man in front of the tool.

Scientific management calls for an intensive standardization of men and machines. As regards the standardization of men, one writer thus comments:

"How can you standardize the strength and character of one man's arm behind the rammer with another?

"How can you standardize the clear judgment necessary in certain manufacturing operations?

"How can you standardize the various intellect and moral character of the men whom one encounters in shop management?"

ECONOMY AND PROGRESSION THE NATIONAL WATCHWORD.

The business and financial interests of the country are fully cognizant of the fact that this Nation has now reached a stage where progressive methods must be installed and rigid economy practiced in order to meet foreign competition. These industrial leaders, however, are also appreciative of the fact that any effort to resort to extremes along such lines will result in

harm rather than in benefit to industrial conditions. The line of action that the most thoughtful are working along is in the direction of that suggested by President Taft, in his address at Portland, Oregon, wherein he stated that " A progressive is one who recognizes existing and concrete evils and is in favor of practical and definite steps to eradicate them." Officials who are directing progress along such practical lines are neither timid nor conservative, but rather the highest and best types of progressive leaders.

Those who doubt the value of scientific management as outlined by Mr. Taylor are not reactionists. It is strikingly noticeable that the special industries which have been able to export their products to foreign countries are, without exception, industries which have had nothing to do with the system. Surely their shop methods and business management must be of efficient character to have their product successfully enter the entire industrial field of the world against the cheaper labor of their European and Asiatic competitors.

INDUSTRIAL MANAGEMENT AN ART RATHER THAN A SCIENCE.

Science has been defined as knowledge reduced to a *system,* and art as knowledge reduced to *practice.*

Industrial and railway management deal with too many unknown and variable quantities to be regarded as a science. The administrative officials of such organizations are well content to reduce their knowledge, experimental research and experience to certain flexible practices and customs, and as a rule the fewer the fixed practices and regulations the greater the efficiency.

The efficiency experts accord too much importance to system and too little weight to practice. As a result the principles of management offered by them are of such detailed, complicated, rigid and expensive nature as to be inapplicable for every-day shop needs. The installation of the whole machinery of scientific management has been very seldom at-

tempted, and it is the rare exception where such installation has been permitted to continue for even a few years.

Some of the problems that these experts recommend to be worked out in individual establishments should only be attempted under commonwealth or national agencies, and then only under conditions where such experimental efforts would not interfere with regular current work.

NORMAL OUTPUT AN AID TO EFFICIENCY.

Probably no factor counts higher for efficiency and economy of output than that of a constant normal production. Where the output is of a constant normal character, substantially accurate data as regards cost accounting can be obtained. Friction between employer and employés ought then be reduced to a minimum, for when labor is assured of steady employment it puts a personality into its work that makes for efficiency, economy and endurance.

Probably the great majority of plants that are operated under constant normal conditions are not of large size, since competition is too keen to permit great industries to be operated under such satisfactory conditions. Where normal output is the rule confidence generally exists between the executive officials and the employés. Substantially any system of management can be attempted at such plants with the assured expectation that it will be given a fair trial.

There are hundreds of plants operated under constant normal conditions, and in many cases they are the principal if not the only industrial establishments located in the vicinity. Under such conditions labor is exceedingly yielding if not loyal to management, and it would appear as if such plants offered ideal conditions for the installation of scientific efficiency methods. It is exceedingly doubtful if any scientific system of management has ever been put in operation or even attempted in a dozen of these establishments, even though it appears to be logically adaptable for such industries, and surely this testimony ought to be of value in showing how

exceedingly inapplicable such system appears for general repair plants, shipbuilding yards and naval stations.

If manufacturing plants, therefore, which turn out a constant normal output of standard articles do not find the system applicable to their needs, wherein is the justification for its installation at a navy yard where the work is not only of repair nature but likewise varies in extent and character? The military engineering needs of the fleet should be the determining factors in the permanent organization of our navy yards, and efficiency results should be regarded as of more importance than stop-watch and time-card methods.

The measure of efficiency for the navy yard for the day of battle should best be determined by the character, endurance and rapidity of repairs and not by the cost as gathered from time cards whose accuracy may even be questioned.

EXTENT OF DEVELOPMENT OF THE SYSTEM.

The glittering generalities and the catching epigrams that have been so diplomatically and successfully used in telling of the advantages resulting from the operation of the system have undoubtedly caused quite a considerable number of plants to attempt to follow, at least in part, the methods inaugurated by Mr. Taylor. The system has therefore been introduced into probably a sufficiently large number of shops of diversified character to thoroughly test its value. The general worth of the system ought to be fairly definitely determined by this time by the industrial managers of the country.

From the best information obtainable the system has been adopted in its entirety by only a very few firms. In a considerable number of cases it has been a simplification of the system that has permitted it to remain in existence. The general opposition and resentment shown by the great majority of shops which have tested the system best tell of its unsatisfactory character.

The system does not wear well, and for this reason only a proportionately small number of the plants which have tried it

are still using it. The cost of installation generally, if not invariably, far exceeds the estimate, and this is due to the fact that the efficiency experts are often without practical knowledge of the special industry whose methods they attempt to revise. As one manufacturer stated, once you permit such experts to attempt to revise shop and management methods, it will be found exceedingly difficult to get them out of the plant, since their professional pride if not their conscience prompts them to remain on the job until they can develop or perfect some system that will compare, at least passably, with the organization that they propose to supplant.

After months of careful inquiry the writer has been unable to learn the names of even a dozen firms that use the system in its entirety. He has not heard that even a score of firms have adopted the substantial features of the project. When pressed for the names of particular establishments that have successfully and continually used the system, there is a disinclination to dwell upon this feature of the proposition. One is thus reminded of the story of Uncle Mose, a plantation negro, who upon being interrogated about his religious affiliations replied, "I's a preacher." "Do you mean," asked the astonished questioner, "that you preach the Gospel?" Mose felt himself getting into deep water, and said, "No, Sah; ah touches that subject very light."

It is reasonable to presume that if the introduction of the system had met with a fair degree of success, the public would have special knowledge of the fact. Prospective purchasers or students of the system are generally advised to visit two industrial establishments in Philadelphia where it is said the scheme has been in successful operation for years. It is somewhat surprising that such success has not inspired a few of the hundreds of neighboring industrial concerns to adopt the essential principles of the scheme. It is only because success is seldom achieved that the scientific managers are reluctant to tell in detail of their accomplishments at other establishments. In turn, the average industrial manager is generally too humiliated to dwell upon the details of the character of

the "gold brick" that he purchased. It is always good business policy to wipe out regret and reference to past failures.

As regards the practical effect of the system upon industrial progress, Mr. Calder thus writes:

"It appeared at first with a more modest title and made its appeal through the ordinary professional channels to the engineer. It was a worthy appeal based upon a quite unusual amount of self-denying investigation, but it did not receive the immediate consideration it deserved. This was partly because the straw man which it set up and repeatedly and vigorously knocked down was merely a lay figure and not really representative, as alleged, of the best existing shop practice.

"In the case of the more open minded and thoughtful engineers, ready to learn from any source, the 'science' of the movement was accepted with considerable reservation, and from the humanitarian point of view the illustrations used by the gifted author of the system laid it open to not unjustifiable attack and to the complaint that though a deeply interesting experiment had been made, it did not justify the far-reaching generalizations based upon it.

"'Science management' itself has caught the fancy of the press and of the man in the street and has been let loose through a popular propaganda upon an indiscriminating public. It will come back to its moorings after a while.

"Actually the particular system described and advocated by Mr. Fred W. Taylor has made relatively little progress and, while economic administration of industrial establishments has been quickened not a little by its advent and discussion, the most of the general advance has been the result of causes operating before that event and much of it has not been along the specific lines of such proposals in 'Scientific Management' as are original with its author.

"The fact of the matter is that Mr. Taylor's 'Scientific Management' is a very big and difficult task requiring professional ability of the highest order. Stripped of the data, apparatus and phraseology which have led careless readers to think of it as a new way of running machinery, of paying men,

of avoiding labor trouble, of insuring dividends, etc., it is neither more nor less in its essence than a proposal to revolutionize our industrial life."

VITALITY OF AMERICAN ARTISAN ALREADY STRAINED.

In a pamphlet issued by the London Board of Trade regarding the industrial conditions of the States of New York, New Jersey and Connecticut for 1910, there is contained the following extract from a report submitted by the British Consul-General of the port of New York:

"Every worker in America puts more energy in his work than does the European in his own country. Speeding is partly responsible for this, but the reserve of energy is no greater in American than in European stock. American energy is consequently exhausted more rapidly. Between the ages of 40 and 50, when the European workman is at his best, the American frequently breaks down. Physical exhaustion, dyspepsia or nervous prostration follows and the man's life as a worker is done. His place is taken by a younger man.

"So long as there is an abundant supply of labor through foreign immigration the vacant places can easily be filled. If the stream stops there will not be so much heard of the superiority of the American working man, for America would then have to depend upon her own children, whose stock of vitality is not greater than that of their parents, whether American or foreign."

Brighter and more energizing conditions of work are required. The daily-task idea is simply another step to enervating work and exhausted vitality, and it is not surprising that welfare experts believe that a halt should be called in the attempt to increase the work pace.

The minute subdivision of operations and the speeding-up of machines constitute two of the basic principles of scientific management. Such practice does not make for the training of high-grade artisans, although it may result in financial benefit to highly-skilled or experienced individual employés.

It is not difficult to train common labor and boys to operate some special machines with rapidity and skill. Such training, however, is neither of permanent benefit to the nation nor the individual, for after the laborer is detailed to such work, it will be the exception when he will cheerfully go back to manual labor. Time-study methods may increase, for a time, the output of the machine, but only at the expense of the development of the man.

The most serious evil connected with automatic work is due to the fact that it makes for mental retrogression of the employé. It is likewise responsible in considerable part for the general socializing process that is going on even in the better class of shops of the country.

In relation to this special feature, Dean Herman Schneider of the University of Cincinnati gave an address on "An Analysis of Work," before the Chicago Commercial Club, on November 11th, 1911. This address contained the results of his investigations of the psychological effect of various kinds of work upon the operator. It was his opinion that when the work became more and more automatic the minds of the artisans became more and more lethargic.

The best kind of citizenship cannot be built on fifty-four hours per week of automatic work, compensated for on a differential basis. The repeated spontaneous and unaccountable strikes of automatic workers bears testimony to the fact that these employés undoubtedly labor under intense strain, and the whole plan of scientific management, at least as regards shop management, is in the direction of increasing both the toil and the strain.

THE RESENTMENT OF SKILLED LABOR TO THE PRINCIPLES OF THE SYSTEM.

In general, organized labor opposes the introduction of the principles of scientific management. In analyzing the cause of this resentment it appears as if the opposition of skilled labor is based upon the following distinct objections:

1st. According to Mr. Taylor's own statement it does not

increase wages in proportion to the increased output produced by the employé.

2d. The employés bitterly resent the implication that they do not render either efficient or conscientious service, and that "soldiering" is the rule and not the exception when day work is carried on.

3d. The labor leaders regard the movement as a step to general piece-work payment—a system which the average artisan believes to be of some benefit to the highly-skilled workman when first introduced, but of eventual detriment to the average mechanic. It appears to be their belief that the compensation for piece work is generally reduced just as soon as the employer finds out the maximum output which can be secured from the high-grade man. It has been asserted by a welfare labor organizer that "The bait of higher earnings for the energetic has too often proven a delusion and snare. Pace making, speeding up and rate cutting are processes with which labor has been long familiar. And, taught by past experience, labor is not unnaturally suspicious now lest scientific management may turn out to be a new method of scientific skinning."

4th. The system has a tendency to entail such an intense and close application to work as to be injurious to the permanent health of the employé.

Rightfully or wrongly, labor thus considers that scientific management is but an attempt to hold it permanently responsible for our business ills and for the general results of inefficient management. The inference may not be founded upon facts, but such appears to be the belief of a very large contingent of both skilled and unskilled labor.

SCIENTIFIC MANAGEMENT IN TEXTILE INDUSTRY ESTABLISHMENTS.

Due to the exceedingly large number of textile workers employed in the cotton, silk and woolen mills, the advocates of scientific management have made strenuous efforts to induce these industries to adopt the system. The success achieved has

not been encouraging, since neither mill owner nor operatives look with favor upon the proposition. The pace work already in existence in the textile mills is about as severe as capital can expect or as labor will endure.

In reference to this matter a very distinguished authority upon cotton manufacture writes: "I am free to state that so far as I can learn from observation whenever opportunity occurred, during the past five years, and from conference with a great many manufacturers, the so-called scientific efficiency system has no place in cotton manufacturing."

One of the scientific experts was a guest of the Cotton Manufacturers' Association of New England, where he read a paper on Scientific Management. Later, at a meeting of the A. S. M. E., he stated that he found comparatively few of the members of the New England Association who were familiar with the details of their shops, and that they were more skilled as merchants than as manufacturers. At a subsequent meeting of the Cotton Manufacturers' Association a vote of protest against what was termed "the inaccurate opinions" of the expert was passed unanimously.

It would seem but natural that the men who have successfully developed great industries from small beginnings should be resentful of both the criticisms and unasked advice of inexperienced experts who possess neither the training nor the facilities intelligently to pass judgment upon the conduct of enterprises that are the outcome of thoughtful study, patient investigation, costly research and extended experience in the business.

Capital, however, upon this question, is compelled to take much more conservative action than organized labor. In particular establishments some wasteful methods may undoubtedly prevail, and therefore no one is justified in unequivocally rejecting any suggestion to effect improvement. There is not, however, an unnecessary wasteful method prevailing that the ordinary manager is not trying to reduce or eradicate, but waste is sometimes a necessary concomitant in the manufacture of special articles, or in carrying on particular processes.

Due to the progressive reduction in the number of orders in the past few years for manufactured products, a considerable number of the most energetic members of the National Metal Trades Association retained efficiency engineers to suggest improved methods of business organization and shop management. Within the past year about fifteen Cleveland companies belonging to one of these Associations thus engaged outside experts to aid them in developing plans for the more scientific management of their shops. In regard to the success achieved by these scientific experts, the annual report of the Cleveland branch of the National Metal Trades Association thus states:

"In the past year, at the request of some of our members, we made an investigation regarding the employment of efficiency engineers in the metal-working establishments in Cleveland. I refer to the employment of outside experts to rectify or remedy so-called existing ailments or to lessen production costs, standardized methods of production and matters of a similar nature. To our surprise we learned that from a great number of experts so employed by different factories in Cleveland the results have not been what were anticipated. The result of our investigation cannot be better expressed than by making extract from a letter from a manufacturer, this being one of many received:

"'We beg to state that our experience with outside efficiency engineers has been on the whole unsatisfactory, and we are impressed that an efficiency campaign or program should not be undertaken in the average factory, except with efficiency men who are permanent members of the factory organization. In the case of our own business we have entrusted all efficiency work to members of our permanently employed staff. The results seem to indicate that this is by far the better way to promote efficiency measures.'"

VIEWS OF AN ECONOMIC EXPERT CONCERNING THE SYSTEM.

Probably no book of reference that has been written concerning the principles of industrial management shows such exceeding thought and investigation of the question as the

work of Professor John C. Duncan, of the University of Illinois.

In analyzing the three kinds of industrial organizations that have been evolved—the military, functional and departmental types—Professor Duncan thus writes of the functional or Taylor system ("The Principles of Industrial Management," pages 191–192):

"Notwithstanding all its advantages the functional system of organization has not proven popular or successful in a number of plants where it has been tried. It causes men to lose initiative. It has a tendency to shift and divide the responsibility in spite of the contrary intention. This has been found to be true in several places where the plan has been tried. The difficulties that have been encountered in carrying the scheme through are:

"1. It requires a great amount of clerical work to fill out instruction cards and write out all orders and minute instructions necessary for the complete enforcement of the scheme.

"2. It is exceedingly hard at times to define clearly to whom certain functions belong and on whom the responsibility rests when things go wrong. For instance, no less than eight bosses outside of the shop disciplinarian come into direct contact with the workmen. Four of these men make out instructions and four others say how they should be carried out. It not infrequently happens that the man who makes out the instructions is somewhat vague in his directions, in the hope that the speed boss or gang boss will make up deficiencies. If a mistake occurs under these conditions it becomes a difficult matter to determine who is to blame, because the instructions man will plead that they were not interpreted correctly and the other bosses will assert that such interpretations could be made. Sometimes the instruction-card man will give instructions and the gang bosses may see a better method. If they do, the chances are that they will want to put their scheme into operation. Hence there will be a conflict of authority. If a boss adheres to the system and doesn't follow the best method possible under the circumstances, the firm is paying for a system

of management which is failing in its purpose of getting the goods out in the cheapest possible manner.

"3. It is cumbersome and expensive to operate. In every shop the jobs must be assigned to men at all kinds of odd times during the day. If a workman desires to start on his job he must come into contact with at least three of those bosses before he can do anything. There are usually several men desiring jobs at one time. Under a system where the workman is supposed to know how to set up a job and interpret instructions, he merely needs to find out what he is supposed to do, and do it, calling on the boss only when there are complications. With this functional scheme he is not supposed to act on his own initiative. As a matter of fact, so many bosses really hinder the work. They irritate the men and are expensive to keep up, because in a large shop they must have a number of sets of bosses to carry out the scheme as laid down."

IMPORTANCE ATTACHED TO THE SYSTEM BY THE MECHANICAL ENGINEERS OF THE COUNTRY.

Probably no better estimate of the general value attached by the mechanical engineers of the country to the subject of scientific management can be shown than by having Mr. Frederick W. Taylor tell of the action of the American Society of Mechanical Engineers in regard to his monograph upon "The Principles of Scientific Management."

It may be incidentally stated that Mr. Taylor is a Past President of the Society. His inaugural address upon assuming that position is an engineering classic and represents years of extended study, heavy financial expenditure and thoughtful research. The various annual proceedings of the Society contain many of his carefully prepared contributions. It could therefore be expected that, upon any subject of particular importance, any article submitted by him would receive special consideration. As regards the consideration accorded his latest work upon scientific management, Mr. Taylor's preface reads as follows:

"This paper was presented by the writer to the American Society of Mechanical Engineers during the month of January, 1910, and has been in the hands of the Meetings Committee for nearly a year. The general interest which has been awakened during the last few months in scientific management has caused the editors of a number of the monthly magazines to visit the writer for the purpose of obtaining materials for articles upon scientific management, to be printed in their various publications.

"The writer has naturally explained the underlying principles of scientific management to these men, in very much the same manner in which he has endeavored to set them forth in the accompanying paper. It is likely, therefore, that several of the articles which will be published will use the same reasoning and very much the same illustrations as those given in this paper.

"Inasmuch as the American Society of Mechanical Engineers has received and published in the past all the early articles which appeared upon this subject, it seems appropriate that this paper, which attempts to set forth the underlying principles of this management, should go before the members of the Society before being given to the general public.

"This object can now be attained only by printing this special edition of the paper, because the time remaining before the publication of the magazine articles referred to is so short as to preclude the possibility of printing the paper in the Journal of the Society. The writer has therefore been reluctantly obliged to withdraw his paper from the hands of the Meetings Committee of the American Society of Mechanical Engineers, and presents it to the membership of the Society with the hope that many of our members may find it of interest."

ATTITUDE OF THE TECHNICAL JOURNALS.

With the exception of a very few of the most influential and important engineering magazines the attitude of the technical press is generally agnostic if not hostile to the system.

This opinion is not only reflected in the letters of various contributors to the various scientific publications but likewise in the editorial columns.

The following editorial published in the "American Machinist" of December 7, 1911, is perhaps characteristic of the position taken by the technical press:

"*The Abuse of the Words Efficiency and Scientific.*—No word in the English language has been more abused than 'efficiency,' especially during the past year or two. This has reached the point where those whose business it is to improve shop output and conditions, balk at the name 'efficiency engineer,' of which they were once so proud and are using different titles to convey the same meaning.

"Next comes the much maligned 'scientific,' and the two words may well be used sparingly until we forget the base uses to which they have been put, and they once more come to mean what the dictionary makers and time-honored usage intended.

"But the agitation which has been raging will make for good, since, after the spasm is over and we settle down to a realization that all of us will bear improvement, also that the human element cannot successfully be bound by the same ironclad rules as a machine, and will always be something of a variable quantity, we shall get down to a more rational and substantial basis.

"There are few of the systems advocated which do not contain some good points, the main odjections being in most cases that they attempt too much. The great danger is in building up a system that collects data which are never used or not worth what they cost, which puts a barrier between the workers and the executives, and which becomes so top heavy that it falls into disrepute and becomes an object of contempt. The system, like a law, must either be enforced or it is worse than useless and should be taken out of active use.

"Another common fault is in attempting to make the system cover too many departments. While in theory a good system should be logically extended to a limit, it is far better

to confine its application to such departments as prove its practical value.

"Nor must it be forgotten that much so-called efficiency is only apparent and not real. And just as the coal pile or the cost of power is the only real indication of a steam engine's efficiency, so the net bank account is the real test of a system's worth."

THE PERSONAL EQUATION IN BUSINESS MANAGEMENT.

Until about twenty-five years ago the home market of this country afforded such a profitable field for manufacturing interests that there was no impelling demand for strict economy as regards details in general business management. The rapid development of industries in favored portions of the South and West, however, gave warning to certain industries of the Central and New England States that hereafter, in order to retain the valuable and extensive trade that had been theirs for several decades, they would have to reduce profits or effect economies.

Instead of meeting competition, there were in many instances a consolidation of certain industries, the principle argument advanced for such action being the large possible saving that could be effected by eliminating many expenses common to each of the separate plants. Through the elimination of competition it was anticipated that the product could be sold at a higher price, and that a very material reduction in administrative, operating and traveling salesmen's expenses could be effected.

For a time all the anticipated benefits of consolidation appeared to materialize. As the best of the managers and technical experts of the individual concerns had been retained by those controlling the mergers, it was not surprising that the net earnings under the new management were generally of very satisfactory character.

It was not long, however, before it was noticeable that some of the old managers failed to continue to take the special interest in the consolidated business which had been given to

their own individual establishments. With the loss of personal responsibility, pride and initiative there came delay in acting upon important matters, since under the consolidation system many matters affecting policy had, as a rule, to be acted upon by a board of directors or by some special committee.

In the administration of these great trusts it was frequently found an easier matter to consolidate plants than it was to consolidate brains, and as a result the overhead expense increased to a surprising extent. As one expert states, "corporation consolidations tend to produce routine officials from top to bottom." It is because individual incentive was sacrificed too much to system that the business pendulum now appears to be swinging in the direction of personal control and authority rather than along the line of centralized effort.

The tendency therefore appears towards decentralization as regards industrial consolidation, since there can be no enduring efficiency in the installation of any system of management which attempts to provide brains and methods for all subordinates. The theory of corporation consolidation is strikingly similar in certain respects to the basic principles of scientific management, since both attempt to achieve success by obliterating the resourcefulness and the initiative of the subordinate. The zeal and resourcefulness of the individual workman is suppressed when he is compelled to use special tools and follow implicitly a blueprint and routine card of instruction which were probably prepared by some one whose practical knowledge and experience did not compare with that of the skilled artisan. The manager of a branch industrial plant likewise naturally resents receiving detailed instructions from distant superiors who cannot possibly have personal knowledge of local conditions.

CONSERVATISM SHOULD BE OBSERVED IN CHANGE OF MANAGEMENT.

The writer has personally conferred with a considerable number of managers in regard to the advantage of effecting radical and sudden changes in the management of industrial

plants. While it appears to be the unanimous opinion of these experts that progression is desirable and should receive all possible encouragement, they are likewise in accord in considering that it is generally detrimental to efficiency to permit any outside expert to enter an establishment and subordinate its organization and operation to his individual views.

In most cases the intrusion of such experts produces the following demoralizing and baneful effects:

The workmen regard with suspicion the advent of such experts, since they are inclined to believe that his entree is simply a forerunner of action that will result in extensive discharges.

It creates exceeding resentment upon the part of the foreman. The more authority there is given experts the more inclined they seem to insist upon changes that are often of unimportant character. The foremen are naturally irritated and provoked to have their professional efficiency discredited by men who are not experienced in the business.

It encroaches upon the time and patience of the managers to a very marked extent. Due to lack of knowledge and experience of the efficiency specialists, it often becomes necessary for the manager and leading foreman to neglect their regular duties in preventing the experts from making serious mistakes.

It has been probably due to the fact that the scientific managers have been given too much rather than too little authority that their success has been of such limited nature. It is in those establishments where the efficiency experts have not had control, but where they have been simply requested to observe and suggest, that the most enduring and satisfactory results appear to have been obtained.

RESULTING ADVANTAGES FROM THE PROPAGANDA.

Careful study of the question, however, will repay close investigation by all interested in industrial operations. The experience already obtained from the application of the system has probably been of some permanent benefit in promoting

efficiency. The possible development of the proposition, as regards administrative organizations rather than shop methods, affords a most interesting study. The resulting direct and indirect benefits have been of the following nature:

Special corporate interests have been compelled to seek advice from sources outside of their own administrative organizations. There is no doubt but that experienced and skilled experts, if afforded opportunity to make extended inspection of industrial plants, can often make suggestions that would promote efficiency. The average efficiency expert, however, often appears more inclined when making such discovery to use it as a pretext for insisting upon wholesale changes in administrative methods and shop management. The ability of an individual to effect detailed improvements does not, however, logically constitute a reason for placing such an expert in control of matters effecting general policy.

The importance of the engineer as regards executive control has been augmented. The education and professional training of the engineer distinctly fits him for executive duty, and where important executive duties have been entrusted to him there has been resulting efficiency and economy. It has been asserted that one of the impelling causes that has made Germany a great manufacturing nation has been due to the importance attached to the engineer as a factor in administrative management.

The value and importance of management have been enhanced. In opposition to the claim made by many efficiency experts that the adoption of scientific management methods would not only render the efficiency of such business self-perpetuating, but that it would also enable industries to become to a large extent independent of their managers and higher executives, a careful study of the problem brings into even greater prominence the value and importance of efficient leadership. Applicable to this feature of the matter, Mr. Calder, Manager of the Remington Typewriter Works, thus comments:

"No army of clerks mechanically following planning in-

structions, however perfect, can take the place of able engineering administrators and shop assistants under any conceivable work system.

"The human element in system, as well as in organization, is half of the problem, and there is a tendency to too great rigidity in most of the shop systems offered for general application.

"It is not a recommendation for any business system, imported from the outside, but rather the reverse, that it should insist upon absolute conformity in type and details without regard to the problem in hand and the great amount of experience already acquired from it.

"At the outset we must reckon with the fact that the organization, not the system, is the primary consideration.

"This is not the order of precedence prescribed by some professional systematizers, but any other is a mistake.

"The organization must move forward as a harmonious unit. No amount of clever scheming alone will secure this. Herein lies the task and the genius of the organizer of men as distinguished from the mere systemizer of things.

"The modern administrator of industrial establishments is a manager of men rather than of things, and the human factor touches his business on all sides.

"An organization, therefore, cannot go into commission. It must have a strong, resourceful leader and a carefully selected, well-trained, loyal and enthusiastic staff. This will only come through intimate contact with a man, not a mere machine or inanimate system.

"The cold-bloodedness of some of the modern schemes for exploiting the higher human energies is not only repelling—it is a fatal defect."

Labor has been compelled to take note of its own shortcomings. The efficiency experts have dwelt so forcibly upon the general inefficiency prevailing in individual plants, that many administrative officials have been compelled to dispense with some of the humanity that has heretofore existed in shop management and to insist upon a more rigid application to

work upon the part of the employés. The first and last half hours of each day, also the quarter hours before and after dinner, are worth as much to the manufacturer as any other part of the working time. Starting and closing operations on time will probably be more rigidly insisted upon in the future than has been practiced in the past. The leniency and consideration accorded the inefficient and irresponsible employés will likely hereafter be considerably abridged. It is not, therefore, surprising that certain organizations of union labor are particularly resentful of the trend of scientific management as regards shop discipline. As a matter of self protection various administrative officials have found it necessary, in deference to the sentiment prevailing concerning waste and efficiency, to demand increased attention to work upon the part of the individual employé.

Increased consideration has been given the question of economic administration of industrial establishments. Spurred on by the frequent unsparing and unjust criticism directed against the management of our industrial enterprises, the administrative and technical officials have bent their best energies to improve their organization methods so far as consistent with humanitarian and welfare principles. These managers were told that their records were incomplete and unsatisfactory; careful investigation shows that the records which it is often proposed to use as substitutes are useless, because their form is so complicated as to render them inapplicable for the purpose of frequent and rapid reference. As stated by one writer upon the subject, it was to an already progressing and intensively developing shop practice that the question of scientific management was forcibly presented. There is probably not an industrial manager of recognized ability in the country who does not have a full appreciation of the far-reaching importance of increasing efficiency even to a small degree, and who does not realize that important results would ensue by improving the relations between the help, the machines and the processes of manufacture. The development of nearly every machine in every line of industry bears

testimony to the progressive character of the administration, organization and operation of the nation's industries.

THE MILITARY VALUE AND IMPORTANCE OF THE NAVY YARDS.

Due to the progressive increase in size of the modern battleships, it becomes more difficult and expensive, with each succeeding year, to maintain the modern naval fleet in a state of efficiency. The military necessity and the engineering value of both shipbuilding plants and naval stations to national defense have therefore become correspondingly greater with the increase in size of the fleets of the world.

From both financial and military standpoints it is extremely advisable that the modern battleship should be made self-sustaining as regards repairs. By reason, however, of the fact that the modern battleship does not, and cannot, contain the installation of machine tools requisite for handling and fabricating heavy machine parts, it may be that with the increase in size of the fleet the work of the navy yards will not be lessened.

The policy inaugurated by Secretary Meyer of demanding that the large complement now comprising the crew of a modern battleship should do its part in maintaining the vessel in a state of efficiency has been productive of far-reaching and beneficial results. While general instructions had been previously issued to utilize the ship's force in making repairs, it is a well-known fact that one of the great industrial departments of our navy yards not only looked askance upon the proposition, but substantially maintained that assistance of such character was rather a hindrance than a help in producing either economy or efficient results. The action of the Secretary has been particularly advantageous in three directions; it has compelled the commanding officers of certain ships to scrutinize more carefully recommendations for changes and requests for repairs; it has given the complements of the ships a better knowledge of the design, construction and installation of the various appliances; and it has developed a system of inspection of repair work upon the part of the ship's force that in turn

has resulted in work being done in a more expeditious, efficient and economical manner. Without the help of the ship's force there would have been either exasperating delay or excessive expenditures in maintaining the battleships in a state of efficiency.

There will always be a continual demand, upon the part of the battleships, for repairs that are beyond the capacity of the ship's force, and in this respect the navy yards are factors that will always count for much in measuring the naval strength of a nation. It has been stated by certain military experts that the actual naval strength of a nation should be measured by its ability to replace its fleet, as well as by the number of battleships that it possesses. From this standpoint the navy yards and shipbuilding plants are important weapons of defense. Speed, extent and character of naval construction are in many respects Great Britain's most valuable naval assets.

Another measure of naval strength is the relative knowledge possessed by the personnel of the several navies as regards the details of design and operation of the various types of ships comprising its fleets. Any navy which does not provide extended shore service for its officers, whereby such men can observe the building and testing of engineering, ordnance and electrical appliances, will never be efficiently prepared for the day of battle.

The navy yards are therefore more than repair plants; they are necessary practical and important post-graduate schools of instruction for men and officers. Those who have been to sea are therefore the logical ones to be placed in full control of every feature relating to the administration and detailed management of our navy yards. Unless the turret and engine-room officers of our battleships are given occasional tours of shore duty where the opportunity will be offered them to acquire an intimate knowledge of the design, construction, installation and testing of the motive appliances installed on the battleships, it is certain that the ships themselves cannot be brought to that state of efficiency which the country demands and expects.

It would be strange if the industrial organization of our navy yards was not founded upon efficiency methods; particularly when there is taken into account the following conditions that tend to develop efficiency of management and operation:

Every commissioned officer assigned to technical duties is either a graduate of the Naval Academy or one promoted for meritorious conduct and efficient service from warrant rank. The mechanical engineering course at Annapolis is probably not surpassed by the engineering course of any technical college in the country.

Every commissioned officer attached to a modern war ship, by reason of the varied technical duties to which he may be assigned, has an excellent opportunity to specialize along such mechanical lines as appear best adaptable to his talents. The latest battleships are equipped with an installation of steam, hydraulic, pneumatic and electric appliances whose cost will approximate about two million dollars. The supervision of the operation and repair of auxiliaries of such diversity and magnitude should certainly fit the modern naval officer for executive work at our navy yards.

As the Government always maintains a considerable force of inspection officers at the various shipyards where naval vessels are in course of construction, the organization and shop methods of the leading American shipbuilding plants are within the observation and study of a considerable contingent of sea-going commissioned officers.

The official log books and records of the Navy are accessible to its officers; and these records contain information of exceeding value to the designer and builder.

Advance information from abroad concerning naval progress may be obtained by all naval officers by making application for such information to the Director of Naval Intelligence.

The organization and administration of many of the leading foreign dock yards also comes within the observation of certain naval officers.

An inspection of both the merchant vessels and battleships of foreign nations is often possible to our naval officers.

The discipline and training which naval officers receive are of decided benefit in fitting them for executive duty. The complement of a modern battleship now approximates 900 men, and it requires executive leadership of high order to train these men for the varied and important duties which devolve upon them.

The fact that the average tour of duty seldom exceeds three years, permits naval officers to be assigned to a wide range of work. The shop methods of men in various parts of the country, therefore, becomes familiar to them.

The fact should also be kept in mind that on board our war ships there are probably four thousand men who have served a regular apprenticeship at some industrial trade. Many of these men have worked in various progressive shops and are therefore quick to point out any obsolete methods that may exist in the manufacturing establishments of the Government. The career of the modern naval officer is now one of continued mechanical experience and training, and the marked success achieved by those officers who have resigned from the service and taken up executive and technical duties particularly tells of their qualifications for such work. The keen rivalry for naval supremacy now existing between nations is a factor that compels the average naval officer to keep abreast of the times, and, whatever may have been the conditions prevailing in the past there is no branch of the Government where one is forced to progress as in the Navy.

There are but few executive officials of commercial industrial plants who do not consider it a tribute to their patriotism. technical efficiency and business ability to be called upon for advice in regard to promoting military engineering efficiency. It is, therefore, possible for commandants of naval stations to obtain technical information from private shipyards in the vicinity of their commands, that might reluctantly be granted or even positively denied individual parties.

THE EFFICIENCY OF OUR NAVAL STATIONS.

The several navy yards are substantial rivals of each other, and each is keen to obtain the highest state of efficiency. The Navy Department notes very carefully the comparative cost of work as well as the relative efficiency and endurance of repairs made at the several yards, and the naval station that falls backward in efficiency is not kept long in official ignorance of the fact.

The officers who have been assigned to duty at various times at both navy yards and shipbuilding plants unqualifiedly state that the organization of the navy yards will, as a whole, compare favorably with the organization of the private yards. There have been built at the navy yards barges, tugs, colliers, cruisers and battleships, and in every case the character of this construction will compare favorably with the vessels built by contract.

Every navy yard is, and should be, an experimental military plant for the development of engineering and ordnance appliances. The character, rather than the quantity of the product turned out, should be the prime factor in determining the efficiency of the station. Cost is measured not only in dollars and cents but in time, and in the case of military engineering establishments national prestige and safety may be involved in the character and rapidity of repairs made at these plants. Those who may be called upon to fight these battleships, and through whose personal efforts the existing efficiency of the vessels has been secured, are surely the logical ones to whom should be entrusted the responsibility of administration and management of our naval stations.

Probably the distinctive weakness of navy-yard organizations is the lamentably insufficient compensation rendered some of the civilian officials. The wage rate of the laborers and artisans is amply adequate for the service rendered, but the pay of the chief clerks, master mechanics, foremen and leading draftsmen is not commensurate with the duties assigned them. Ever since the close of the Civil War there has been

a progressive increase in the wages of the navy-yard mechanics and the wage rate of the laborers and artisans now probably averages 50 per cent. more than it did thirty years ago. As the compensation of the leading men, however, has not been increased proportionately, it is not surprising that the navy yards do not always tempt the class of men that are essential for the exceedingly important duties of foremen, master mechanics, chief draftsmen and chief clerks. The officials of private shipbuilding plants, performing duties substantially corresponding to those assigned the leading civilian employés at the navy yards, receive compensation far in excess of that paid the navy-yard officials. In some cases the shipbuilding plants pay double the amount for corresponding superintendence.

THE VICKERS (LTD.) SYSTEM.

With a thorough realization of the fact that the navy yards are but a series of mechanical auxiliaries to the fleet, and yet auxiliaries of exceeding military importance, Secretary Meyer, upon assuming office, gave special and extended consideration to the problem as to how the military and industrial efficiency of the navy yards might be augmented. Information was sought from every possible source, as well as from interests having widely divergent views upon the matter. The services of several well known efficiency experts, as well as leading cost accountants were retained.

Numerous boards were appointed to collect data and to submit recommendations, and as a rule every interest in the Navy had representation on such boards.

In his extended and thoughtful study of the problem Secretary Meyer made a careful inspection of the principal dock yards and shipbuilding plants of Great Britain. As shipbuilding is a profitable industry in Great Britain, management should naturally have been developed in that country to an exceptionally satisfactory degree, and therefore the visit to the English dock yards gave the Secretary an excellent opportunity to note the comparative merits of American shipbuilding methods

with those of British practice. It also permitted him to note the trend of advance as regards management of the English shipbuilding plants.

Secretary Meyer was particularly impressed with the administrative organization and management of the Vickers (Ltd.) Shipbuilding Plant. The management of this establishment is based upon the assumption that the organization and work of the three branches of the plant—Hull, Engineering and Finance—are so technically and physically distinct that each division should be managed independently of the other. Instead, therefore, of combining the operation of the three branches under one manager, it is considered that more satisfactory results are secured by entrusting the management of each division to an individual Director. These three Directors form a local Board who are expected to coördinate their work in such manner as will best promote the purpose and policy of the London office. While the Engineering Director, by virtue of his seniority, can exercise authority in any branch of the plant when deemed necessary, it is understood that this prerogative is seldom exercised.

The Vickers System is considered to accomplish the following:

(a) Expeditious estimating.

(b) Speed in getting work actually started in shops.

(c) Absence of copious preliminary paper work.

(d) The "following up" of work by means of alert intelligence (progress men) in lieu of multiple automatic card service.

(e) The checking of locations of components under manufacture, by means of a "stores" clearing house.

(f) The securing of increased effort and initiative on the part of the workmen by means of a well-considered premium bonus system.

Except in respect to difference of detail in management that could be expected to exist in the industrial establishments of different nations, the system at Vickers is strikingly similar in many essentials to that prevailing in some of the leading

industrial plants of America. The system in general is founded upon common sense and experience, and attaches little value to espionage over workmen beyond that essential to efficient supervision. Judged from American practice and experience, it might seem advisable to dispense with some of the progress men detailed under the Vickers system for following-up work and detail these experts to the duties of quartermen and sub-foremen. Much of the work arranged for the progress men would naturally be taken up by an efficient and conscientious supervisor.

As a result of the Navy Department's extended investigation of the matter, there is being installed at the Norfolk Navy Yard a system of management that is in certain essential respects the system in existence at the Vickers Works. As regards one detail of the Vickers system, payment by the premium bonus system, the Navy Department's past experience with piece work does not appear to have been satisfactory. There are probably other minor features of the Vickers System which the Department may not regard as applicable to American industrial life.

The friendly but spirited rivalry existing between the various navy yards undoubtedly will cause all rival navy yards to give the most careful and extended consideration to the system inaugurated at Norfolk. It can reasonably be expected that the officials at the Norfolk Station will make a very determined effort to develop the system so as to make it particularly applicable to the needs of our naval service. The outcome of extended investigation of the matter combined with practical trial and experiment will undoubtedly develop a system of management that ought to promote industrial economy and augment military efficiency. The Navy and the Nation are to be congratulated upon the action of the Secretary in thus directing the installation of a system of management that is not of that inflexible and exasperating nature which appears to be the distinguishing feature in many of the systems designed by the professional scientific managers.

NATIONAL ENDORSEMENT DESIRED.

Coincident with the wholesale condemnation of general business management and operation the professionals have made a most persistent and special effort during the past few years to compel some of the executive departments of the Government to give at least limited official endorsement to the system of scientific management. In fact, the more aggressive exponents of the science are even demanding that a new executive department be created and that such department have not only exclusive control of all duties relating directly or indirectly to engineering and business affairs, but that it be administered along the inflexible and intolerant lines that have in general been productive of subsequent resentment.

It is somewhat difficult to understand in which department of the Government the principles of scientific management could be made applicable except to a very limited extent. The organization of each of these departments represents extended experience and progressive development. Their methods and operations are continually under the supervision of both friendly and hostile critics. Friendly critics naturally are desirous of overcoming all inherent weaknesses as regards organization and management, while hostile critics are bent upon discrediting the existing organization. Progressive development has therefore taken place under the spur of both friendly and hostile critics. Is it not, therefore, reasonable to presume that at least in the manufacturing and technical departments it will be found somewhat difficult to effect any radical change in carrying on the business affairs of the Government, and that any other than conservative and progressive changes would result in detriment to the public good?

While the adoption of scientific management by even a single division of any executive department might not directly affect any considerable number of individuals, the indirect results are likely to be of far-reaching nature. This fact is not only appreciated by the leaders of various labor organizations, by scientific management experts, but likewise by the manufacturing interests.

With any unqualified official endorsement of scientific management by an executive department of the Government there would result a manifold increase in the ranks of scientific managers, business researchers, industrial organizers, efficiency experts and the numerous other technical specialists who are bent upon telling successful business managers of some better way to operate their establishments. The courts would be besieged to assign these experts to the receivership of financially embarrassed firms. There seems to be a particular desire upon the part of the efficiency promoter to develop into a very important concomitant of municipal and national administrative affairs. Some of these professors of efficiency actually expect many of their visions of extended influence to turn into verities. Such may be the case, unless the engineering, educational and commercial associations take some concerted and far-reaching action in showing the extreme limitations of the system as regards industrial matters.

As regards the industrial extension of scientific management, reaction already has set in; and, therefore, it can well be understood why the desire is keen upon the part of the efficiency promoters to stem the tide of commercial opposition and resentment by governmental support and endorsement.

REPORT OF THE GANTT–EMERSON–DAY BOARD ON INDUSTRIAL MANAGEMENT OF UNITED STATES NAVY YARDS.

In examining the report of the experts on Scientific Management one is struck by the fact that the members of the Board realized that they were confronted by conditions which were new to them, conditions that do not obtain in industrial establishments with which they may have been more or less familiar. It is apparent that they did not know how to meet the conditions existing at the navy yards. This is shown all through their report, but best finds expression in the statement that "The demands on the navy yards are large military in character and therefore outside the scope of our experience." Their unfamiliarity with navy-yard demands is also reflected

in the advice to the Secretary to "apply the same thorough methods in the navy yards that you have adopted in the fleet."

These gentlemen had previously to do with *manufacturing* establishments where the same kind of work is done day in and day out, and where they could lay down specific rules for the operation of shops. In the navy yards they found big *repair* shops as distinguished from manufacturing shops, and the magnitude of the job was appalling in the variety of work that these shops were called upon to do. The subject required treatment far different from what they had previously imagined, and in their perplexity they exclaim, "it is only when the number and variety of the operations to be performed in a navy yards are recognized that the size of the undertaking can be realized."

And yet there are those who would rashly and ill advisedly overturn existing organization and install a system of management that was attempted at one or two American shipbuilding plants, but which was so expensive to maintain and which was so exasperating to labor that every vestige of such system has had to be eradicated. There is not a single executive administrative official of a shipbuilding plant who considers the management question of a navy yard as a simple one, and as one that could be handled better by him than by the Secretary of the Navy and the able officers who are coöperating with him. Even the Board stated that "It must be realized, of course, that several years will be necessary to bring about even *an approximation* to this ideal."

Notwithstanding the awe with which the Board approached the question of organization, it made certain specific recommendations which the Secretary of the Navy is already carrying out.

They recommended a "central organization" for navy yards, and this has been carried out under an officer who is known as the "Director of Navy Yards," to whom is assigned, as far as can be under the law, the specific duties enumerated by this Board.

They recommended a planning department, and an efficient

system has already been introduced in each yard. It is not, however, of the inflexible and detailed character recommended by the Board.

They recommended the establishment of a separate corps of specialists, which could not be done; but, realizing this, they recommended as best, some plan which "would develop a more efficient personnel and effect more complete coöperation." The Board also recommended there be "but a single class of officers, which would include all specialists within its ranks." This the Secretary is endeavoring to carry out by legislation aimed to consolidate the naval constructors and the paymasters with the line officers of the Navy.

They also suggested that the plan which they recommended in abstract form be introduced in its entirety in a single yard, and that certain features be introduced in all yards. This is being carried out, but not quite on the plan which these gentlemen had in mind, for it must be remembered that they were all apostles of what has come to be known as the Taylor System, which is so repugnant to our ideas of what is due the mechanics in our navy yards. The prominent feature of this system is "time studies," exemplified by a clerk with a stop watch timing each operation of the workman in order to set a standard which none but the most expert and the most vigorous can reach, a standard which is far above the average performance and upon which the workman's rate of pay is fixed.

It was such a system as this which was being gradually introduced into the navy yards and which was checked by Secretary Meyer when he commenced to investigate conditions, as even these experts state that it would take several years to reach an approximation to ideal conditions, it is not surprising that little was accomplished. Mr. Meyer was not satisfied that such a system would be a success in our navy yards; indeed, he was convinced that it could not be, and, while he was aware that there was room for much improvement, he was not blind to the fact that improvements must be along sane lines and must be carried out under a system which would

not work a hardship on the employés, still less subject them to some of the humiliation which accompanies typical scientific management as embodied in the Taylor System.

That there has been marked improvement of navy-yard conditions during Mr. Meyer's administration is attested by the fact that the Fleet is today the largest in the history of the Navy and that it is being maintained at less cost than it was a few years ago, notwithstanding the fact that the new vessels added to the Fleet are in some cases fully twice as big as those they replaced.

That Mr. Meyer's judgment is correct in not adopting the controlling features of the so-called Taylor system is manifest in the reflection that there is not a single shipbuilding establishment in this country which operates under it.

Mr. Meyer has all along been impressed with the fact that officers who achieve success in handling large bodies of men on board ship will be most successful in like positions on shore when re-inforced by men of exceptional ability in industrial operations. That this opinion was also shared by this Board is evident from the concluding paragraph of their report, which reads

"While navy-yard problems differ in detail from those presented by the operations of the fleet, yet substantially the same possibilities are presented in each case, and a full recognition of this fact, coupled with the subordination of all detail considerations to the end sought, should place the Navy organization in a position where it will be the national example of efficiency."

The one portion of the report of the Board of Experts on Scientific Management which would seem to reflect upon the organization of the navy yards is found on page 19, where it is stated that

"It is impossible to expect efficient management when conditions prevail such as are evidenced by the records from navy yard, Philadelphia, where the tenure of many engineer officers holding important executive positions has been less than a year."

This statement was evidently very carelessly prepared, and it is evident that the Board must have consulted other "records" than the official ones, for the official record shows that only *one* officer of the Machinery Division had been on duty at Philadelphia less than a year, and he the youngest one. Mr. Meyer's plan contemplates giving these young officers experience in navy-yard shops in order that they may become familiar with the methods of doing work and thus be in a position to know how to execute repairs on board ship, and in this way make the Fleet as nearly as possible self-sustaining.

Is it not a fact that only one of the "experts" ever visited the Machinery Division of the Philadelphia navy yard, and that this member spent about two hours in consultation with the Commandant, about one-half hour in the office of the Engineer Officer of the Yard, followed by a hurried trip through the Machinery Division shops and a short visit to the *Idaho* to see the character of the work in her turrets, the entire time devoted to a personal investigation of the machinery division having consumed not more than one hour?

The other engineer officers at Philadelphia had, of course, had much more experience than the youngest one: The one at the head of the Machinery Division, and to whom the others were assistants, is an officer of exceptional ability, having had 37 years' service, every day of it as an engineer, and at the time this report was written had been 3 years at the Philadelphia yard. He had also been head of the engineering department of the Cavite navy yard for several years. It is, therefore, apparent that the affairs of the Machinery Division were in competent hands. At the time of the visit of the "expert" this officer had about worked out a system of management for navy-yard shops which, though not embodying the refinements insisted upon by advocates of the Taylor system, has been pronounced by many officers and master workmen as being admirably adapted for the conditions which navy-yard shops have to meet. Can it be that the slap of the "expert" was intended as disapproval of this officer's system because of his failure to recognize the

Taylor system as being the only one applicable to navy-yard needs?

The principal assistant to the Engineer Officer was an officer of 19 years' service, who at the date of the report had been nearly two years at the Philadelphia navy yard; and previously had served with distinction in the gun factory at Washington, being a recognized expert in ordnance and electricity. The other officers had been at Philadelphia 18 months.

It may not be amiss to compare the organization of the Machinery Division at Philadelphia as it existed at the time of the expert's visit with what it was when the Newberry system went into effect. At that time the present experienced Engineer Officer was in charge of the shops of what is now known as the Machinery Division. As soon as the Newberry system went into effect he was displaced and the work of the Machinery Division distributed between two young assistant naval constructors. The senior of these assistant constructors had only completed his college course 20 months before and for the last 8 months of this time had been supervising hull repairs at Philadelphia. The junior, who was charged with the important duty of making estimates and *recommendations* for machinery work, had had 8 months' navy-yard experience in the hull department, during which time his principal duty was the "supervision of requisitions and the purchase of material." (Report of Secretary of the Navy, 1908, page 512; 1909, page 491.) He had had absolutely no experience either with the operation or the repair of motive machinery, and had completed his course of study at the Massachusetts Institute of Technology just 8 months before he was ordered to Philadelphia.

If the board of "experts" could criticize an organization where the *junior* assistant had had only one year's shop experience, imagine what their criticism would have been if confronted with the organization under the Newberry System where the officer in charge of recommending what repairs to machinery should or should not be undertaken had been only 8 months out of school, had had only 8 months' navy-yard ex-

perience, and none whatever in the special work of which he had been appointed to assume charge.

CONCLUSIONS.

The writer has conversed with numerous technical experts and executive officials in regard to their estimate of the value of scientific management. Their general opinion appears to be reflected in the answer made by a young matron who, replying to the question as to whether her youthful husband was all she expected him to be, stated that, "he was pretty much what she thought he would be, but that was a whole lot short of what he thought himself to be."

Scientific management appears to fall far short, in the following respects, of what its advocates claim:

It is exceedingly costly to install.

It attaches too little value to the humanity of labor. Men cannot be forced like machines.

It discredits labor to an inordinate degree by impeaching its integrity. It attempts to set a daily task of such character as to be beyond the capacity of the ordinary workman.

It represses the resourcefulness and initiative of the individual workman.

Intensive work can never be obtained for any extended period from the average mechanic. In order to promote his own interests or to accomplish some definite purpose the individual, for a brief time, may work under such strain. The intensive demands of the system will, however, always be a bar to its extended use.

It arouses the resentment of shop superintendents and administrative officials by the tactless manner in which it attempts to change and discredit existing shop methods and business management.

It calls for an increased staff of planners and shop foremen, although the labor market is already short of competent quartermen and subforemen.

It exploits system to such a degree that it becomes a hin-

drance rather than a help to efficient management and industrial development.

Experienced, conscientious and highly proficient mechanics will not cheerfully submit to any system of petty tutorage. There are so many good shops open to them where they will not be subjected to the humiliating situation of having every motive impeached and every motion supervised that they will not remain in shops where such methods prevail.

Particularly for repair work the system tends to discourage the mechanic from any inclination to exercise resourcefulness and to devise improvements for permanently overcoming any inherent defects of design or construction. It is a well-known fact that for repair work concentrated effort and sense of personal responsibility often results in effecting important changes of both arrangement and design. The effort to suppress such resourcefulness upon the part of the individual, and the inclination to cast all inventive effort upon management, is not in line with engineering advance; it partakes more of the nature of retrogression.

It attempts to put into force methods of precision, refinement and discipline that are not compatible with practical shop management.

The salient principles of the system have been subjected to such limited application that in the attempt to operate the system for any extended period, it has been found, notably in the case of the shops of the Santa Fe Railroad Company, that the projected methods of operation were of such unstable character as to necessitate constant changes of administration; these changes even extending to the administration of the bonus system of payment.

Spectacular promises and startling deductions seem to be the general rule in exploiting the system and telling of its operation. Actual and permanent results, and not promises and deductions, are the factors which determine real efficiency. In the practical operation of the system it is the exception where the fancies of the enthusiasts have materialized into substantial accomplishments, when measured either from

the standpoint of net profits or the humane operation of mechanical activities.

There is not a single article manufactured under the Taylor scientific management system whose cost has been reduced to the consumer.

The numerous isolated instances cited by various experts wherein efficiency has been augmented are, as stated by "Engineering" (London), "examples of a class of improvement which is being carried on every day and all over the world by people who never heard of motion study."

The proverb "that those who live the longest see the most" appears to be applicable in this case. It is significant that, with the exception of its originator, the movement has not appealed to one recognized captain of industry; nor does it appear to have met the approval of any one identified with the distinct development of any mechanical industry. One has yet to hear of a half-dozen men of mature age, commanding executive ability and of recognized talent, who have recommended adoption of the scheme in anything like its entirety.

Although its advocates claim that the system is simply another application of the moral law of the universe, it seems quite incongruous that the recognized leader of the movement should unqualifiedly assert, that even though the productive power of the workmen should be increased three-fold, the laborer, by reason of his character and temperament, was entitled to but 60 per cent. increase of wages. Such a code of ethics surely falls far short of the basic principles of the moral law.

While the purpose and principles of the system may for a brief period meet with the approval of a certain portion of the general reading public, such endorsement is of that character which commends every movement that has for its object the extension of civic righteousness, public comfort and national efficiency. The deep-seated opposition of organized labor, combined with the skepticism if not hostility of the industrial interests, makes it extremely inadvisable for either municipal, state, or national authorities to give even semi-official approval

to the exploitation of the system. Any official endorsement will likely be distorted or interpreted in a manner that will not reflect the purpose and spirit intended. It will be a bar rather than an aid to promoting either trade conditions or national efficiency.

It has been the combination of patient investigation, systematic engineering training and directive ability of its commissioned personnel that has given the American Navy its distinguishing lead in fire control, turret efficiency and engineering accomplishments. Scientific management as expounded by its advocates would have nullified, in great part, this remarkable development of engineering and ordnance advance.

If measured by the results produced, the existing management of our navy yards ought to be regarded as highly satisfactory. Their efficiency has been shown by the satisfactory performances of the battleships that have been repaired at the various naval stations.

The continued control and detailed management of the navy yards by the commissioned personnel is of the utmost importance to the efficiency of the fleet and to the good of the Nation. These naval stations are more than important military adjuncts to the fleet; they are, for the great majority of the commissioned personnel, the only post-graduate schools of instruction available whereby these officers are enabled to obtain that detail knowledge of the design, manufacture and installation of naval appliances which must be acquired by those to whom the honor and safety of the Nation is entrusted on the high seas.

THE NAVY'S INTEREST IN THE CAPE COD CANAL.

By J. W. Miller, Late Lieut. Comdr. U. S. Navy.

The need of a canal through the eight-mile isthmus separating Barnstable from Buzzard's Bay has long been felt. The Government, the State and individual effort having failed to build it, a Massachusetts corporation is now constructing it.

From the thirty-foot depth at either end the distance is thirteen miles. It runs through the narrow Bourne valley where the maximum cut is only twenty-nine feet above sea level. The depth of the channel will be twenty-five feet at low water. There will be no locks. The difference in the tidal level between the two bays, being about five feet, will give a current for a period of about half an hour during spring tides of three knots. The soil is more stable than was anticipated and can be relied upon to prevent attrition, although a revetment of granite will be provided to overcome any possible "wash" from vessels steaming at a six-knot speed.

The bottom width is to be one hundred feet, and in the central portion, where there will be two passing points, two hundred feet. That of the approaches varies from two hundred to three hundred feet. One railroad lift Bascule bridge, and two similar, though smaller ones, cross the canal. These can be opened in one minute.

The canal and the approaches will be illuminated on the street-lighting electric system and gongs will be provided as fog signals. Thus vessels will be able to step from one aid to navigation to the next and will avoid overlapping sound and the blinding of pilots through far-reaching rays.

A substantial breakwater 3,000 feet long, containing 400,000 tons of granite, amply protects the eastern entrance from the

northeastern gales. The experience of the past two severe winters proves that ice will not retard steamer navigation, and definitely shows that the drift of coast sand is a negligible quantity.

From a practical navigating point of view the canal should be more perfect than any at home or abroad. Financially it will be remunerative. The securities of the Boston, Cape Cod & New York Canal Co. consist of $6,000,000 five per cent. bonds, and a similar amount of stock. The traffic today passing through Vineyard Sound is 25,000,000 tons per annum. The canal will save seventy miles of distance between Nova Scotia and our Southern Atlantic coast ports. It will shorten the average trip two days in time, lessen insurance rates, and diminish fog dangers forty per cent. Vessels will be able to make about fifteen more trips per annum.

Computing the known tonnage against the interest and a proper dividend on the stock, a result is easily reached showing that the tolls can be made attractive to commerce and gainful to the company, especially in these days when quick passages are essential even for crude material, such as coal, of which there are 11,000,000 tons carried past Cape Cod each year.

II.

Such, in brief, are the general features of the Cape Cod Canal and their relation to coastwise commerce. To the naval student the question of financial returns scarcely appeals. To him comes the direct questions: How will the canal affect the service? and, Will it contribute in any way to defense or to the increase of sea power?

These questions are worthy of attention today when laymen throughout the country are awakening to a renewed interest in "improved waterways." This increasing interest, starting from provincial or selfish motives, is gradually concentrating upon the broad principle of ocean outlets from the larger navigable rivers, as well as upon improved coastwise channels. These latter built, the passage of men-of-war and torpedo

CHART

SHOWING

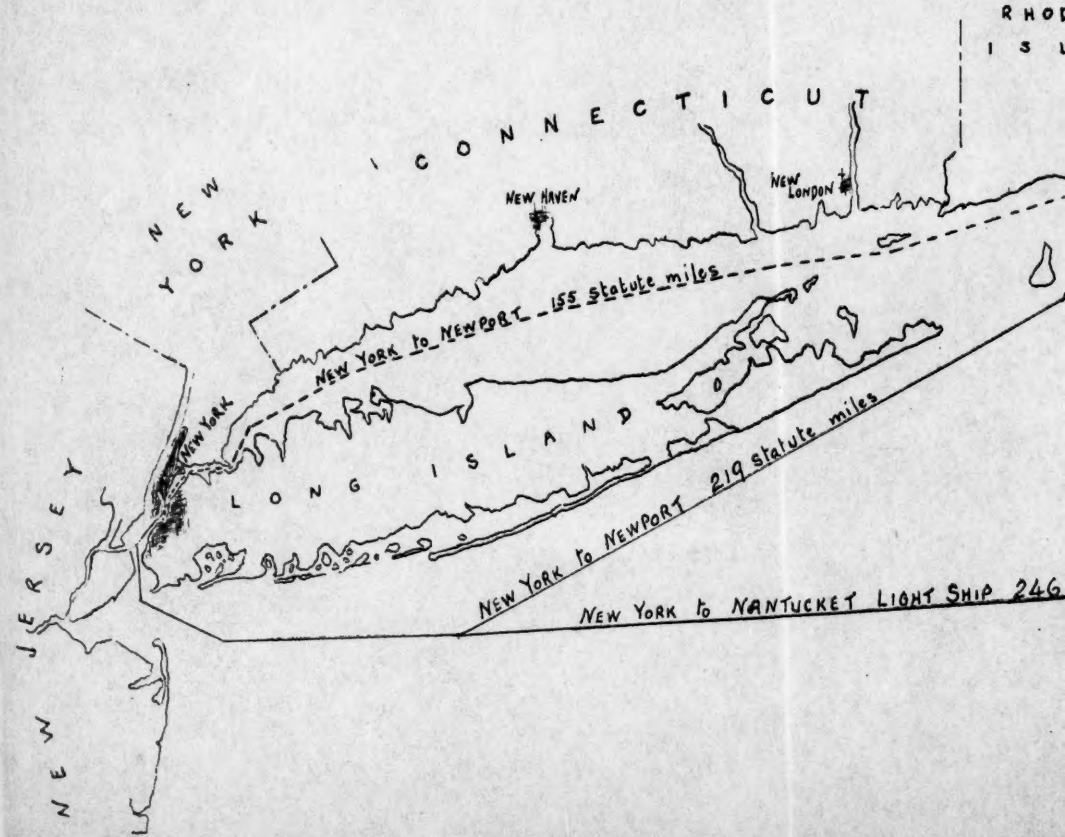
Existing and proposed battleship routes

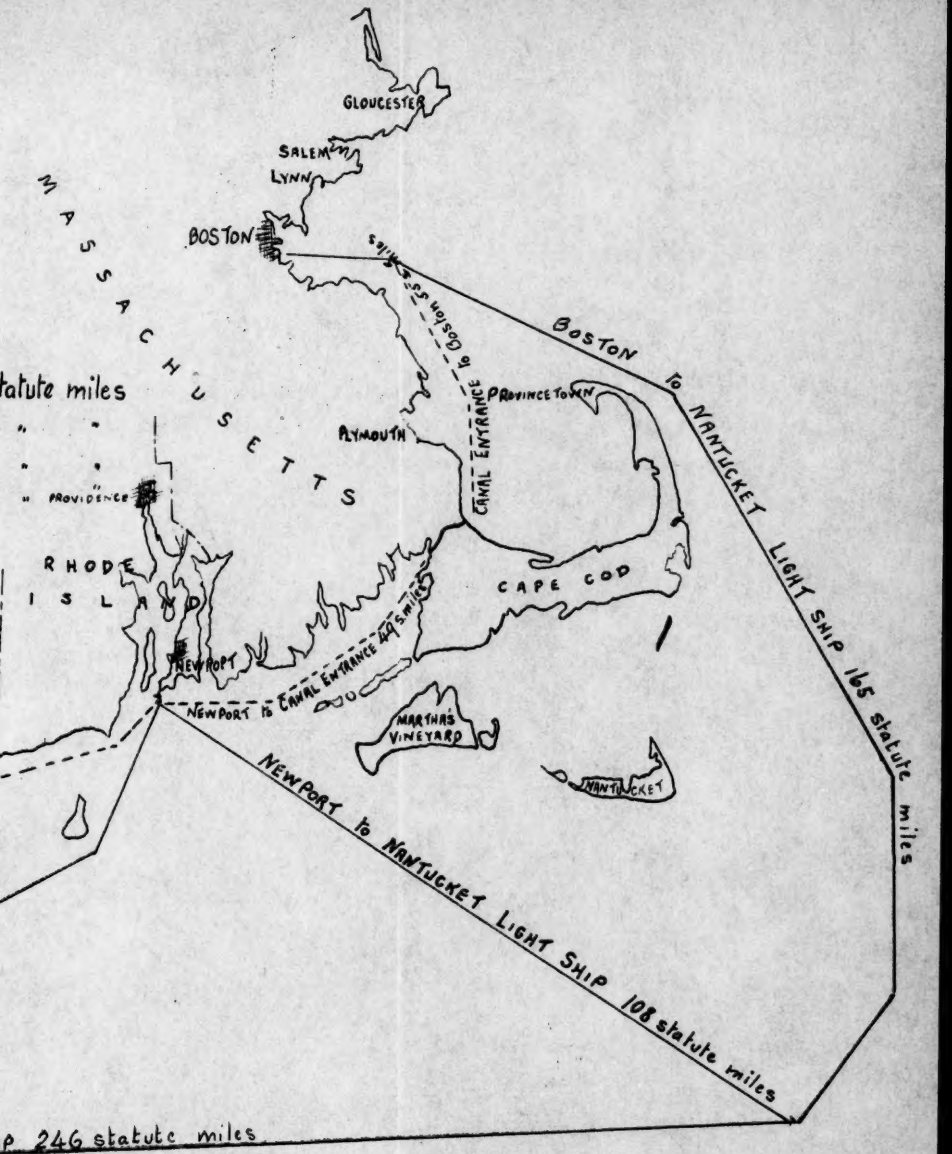
BETWEEN

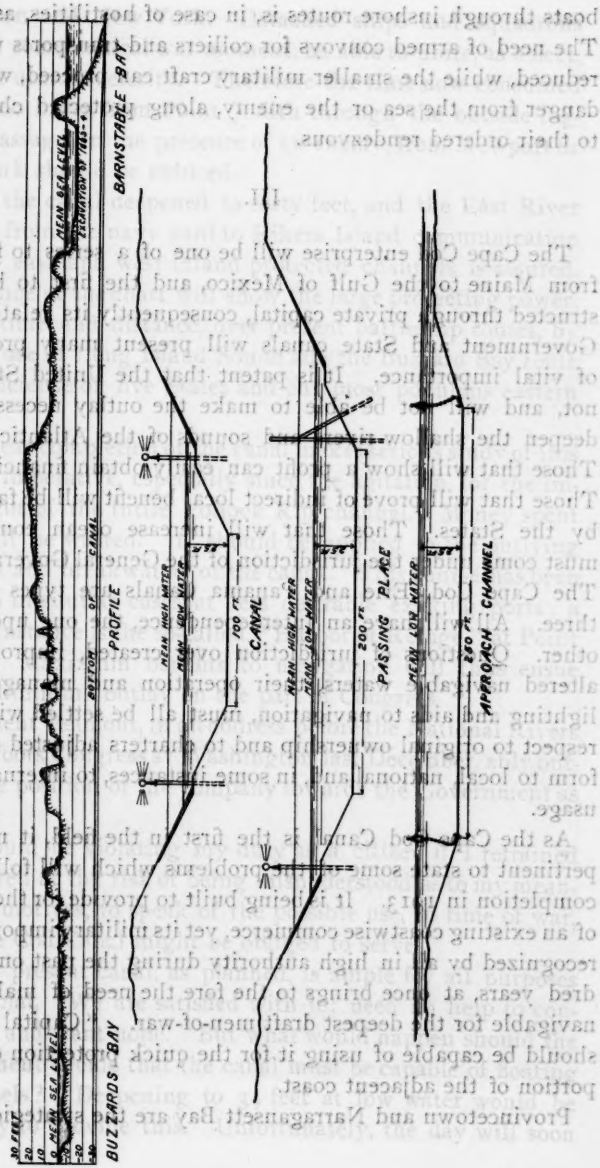
NEW YORK AND BOSTON

DISTANCES

New York to Boston via Long Island Sound and Cape Cod Canal	267	statute miles
New York to Boston outside Nantucket Lightship	411	" "
Newport to Boston via Cape Cod Canal	112	" "
Newport to Boston outside Nantucket Lightship	273	" "




GLOUCESTER
SALEM
LYNN
BOSTON
MASSACHUSETTS
statute miles
" "
" "
" PROVIDENCE
RHODE ISLAND
PLYMOUTH
Boston 45 miles
CANAL ENTRANCE
PROVINCE TOWN
BOSTON to NANTUCKET LIGHT SHIP 165 statute miles
CAPE COD
NEWPORT
NEWPORT to CANAL ENTRANCE 149 miles
MARTHA'S VINEYARD
NANTUCKET
NEWPORT to NANTUCKET LIGHT SHIP 108 statute miles
p 246 statute miles


BARNSTABLE BAY
MEAN SEA LEVEL
ESTIMATED TIDE
BOTTOM OF CANAL
PROFILE
MEAN HIGH WATER
MEAN LOW WATER
100 Ft.
CANAL
MEAN HIGH WATER
MEAN LOW WATER
200 Ft.
PASSING PLACE
MEAN LOW WATER
250 Ft.
APPROACH CHANNEL
BUZZARDS BAY
MEAN SEA LEVEL
10
0
10
20
30

boats through inshore routes is, in case of hostilities, assured. The need of armed convoys for colliers and transports will be reduced, while the smaller military craft can proceed, without danger from the sea or the enemy, along protected channels to their ordered rendezvous.

III.

The Cape Cod enterprise will be one of a series to follow, from Maine to the Gulf of Mexico, and the first to be constructed through private capital, consequently its relations to Government and State canals will present many problems of vital importance. It is patent that the United States is not, and will not be able to make the outlay necessary to deepen the shallow rivers and sounds of the Atlantic coast. Those that will show a profit can easily obtain financial aid. Those that will prove of indirect local benefit will be fathered by the States. Those that will increase ocean commerce must come under the jurisdiction of the General Government. The Cape Cod, Erie and Panama Canals are types of the three. All will have an interdependence, the one upon the other. Questions of jurisdiction over created, improved or altered navigable waters, their operation and management, lighting and aids to navigation, must all be settled with due respect to original ownership and to charters adjusted to conform to local, national and, in some instances, to international usage.

As the Cape Cod Canal is the first in the field, it may be pertinent to state some of the problems which will follow its completion in 1913. It is being built to provide for the needs of an existing coastwise commerce, yet its military importance, recognized by all in high authority during the past one hundred years, at once brings to the fore the need of making it navigable for the deepest draft men-of-war. "Capital ships" should be capable of using it for the quick protection of any portion of the adjacent coast.

Provincetown and Narragansett Bay are the strategic bases

for Boston and New York. Detached ships and squadrons must, in some critical war moment, be able to unite, as a fleet, with the utmost celerity. Therefore the time now consumed in the 160 miles from Point Judith through the outside fog-bound passage, in the presence of an enemy, from Newport or New York should be reduced.

With the canal deepened to forty feet, and the East River dredged from the navy yard to Rikers Island communication through east and west inland protected channels is assured.

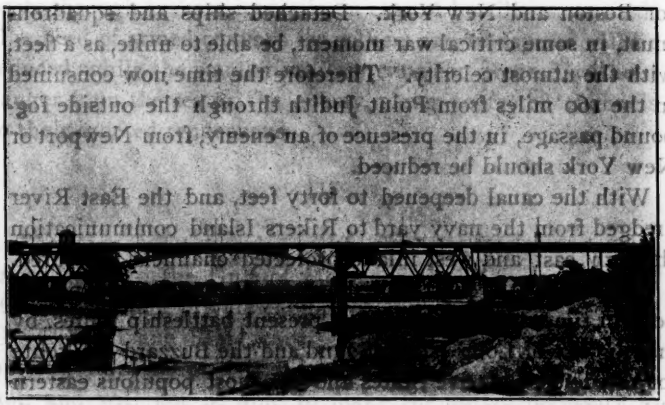
A glance at the chart will show the large protecting power, both in time and distance, over present battleship routes, by making use of Long Island Sound and the Buzzard Bay route for the defense of five States and our most populous eastern cities.

The near completion of the canal makes serious study of this subject imperative, especially since the agitation for the improvement of the futile Pollock Rip channel. Money spent there will be wasted. It should be applied to the outlying channels and breakwaters of the canal. This course has been followed in similar cases at less desirable existing ports; a notable instance is the so-called "Harbor of Refuge," at Point Judith. Maximum benefits to navigation will thus ensue from a minimum outlay on the part of Congress.

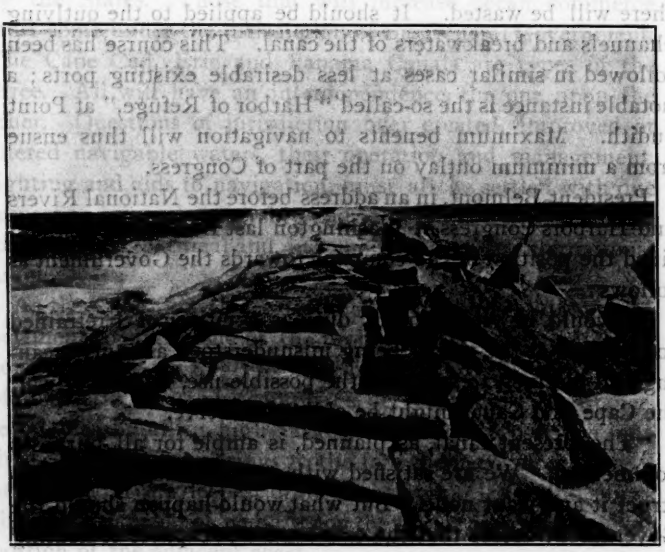
President Belmont, in an address before the National Rivers and Harbors Congress at Washington last December, ably outlined the position of the company towards the Government as follows:

"I would be violating my duty as a citizen if I refrained today, even at the risk of being misunderstood as to my meaning and motives, to speak of the possible use, in time of war, the Cape Cod Canal might be obliged to serve.

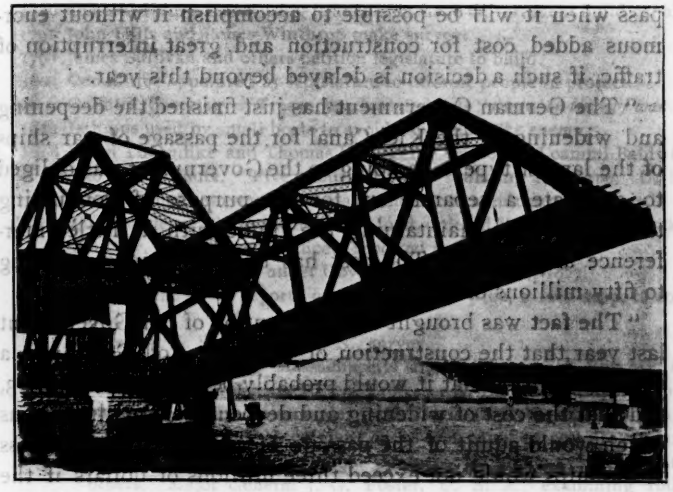
"The present canal, as planned, is ample for all purposes commercial. We are satisfied with it; need no help to construct it and want none. But what would happen should the Government decide that the canal must be capable of floating war vessels? Deepening to 34 feet at low water would be necessary to provide this. Unfortunately, the day will soon



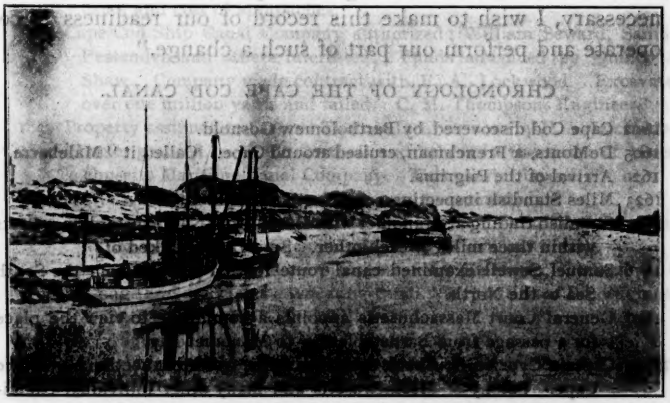
SCHERZER DOUBLE-LIFT HIGHWAY BRIDGE AT BOURNE.



BREAKWATER, BARNSTABLE BAY, SHOWING SMOOTH CHANNEL, LEE SIDE, DURING NORTHEAST STORM OF JANUARY 6, 1912.



RAISING STRAUSS SINGLE-LIFT RAILROAD BRIDGE (WEIGHING 2,000 TONS) AT BUZZARDS BAY IN 58 SECONDS.



THREE-MILE COMPLETED CHANNEL, SAGAMORE TO BARNSTABLE BAY, SHOWING NO INTERRUPTION OF NAVIGATION FOR TUGS DURING WORST WINTER WEATHER.

pass when it will be possible to accomplish it without enormous added cost for construction and great interruption of traffic, if such a decision is delayed beyond this year.

"The German Government has just finished the deepening and widening of the Kiel Canal for the passage of war ships of the largest type. In doing so the Government was obliged to excavate a separate cut for the purpose of conducting the work while maintaining the traffic with as little interference as possible. The cost has been enormous, reaching to fifty millions of dollars.

"The fact was brought to the attention of our Government last year that the construction of the Cape Cod Canal was a definite reality; that it would probably be completed by 1913, and that the cost of widening and deepening it to dimensions which would admit of the passage of our Dreadnought-class of warships would not exceed three millions of dollars if the decision was reached and the work of widening begun before the spring of 1912. I have heard nothing further on the subject and presume the use of the canal for such a purpose is not considered advisable by our best authorities in such matters. Should time change this opinion and action become necessary, I wish to make this record of our readiness to cooperate and perform our part of such a change."

CHRONOLOGY OF THE CAPE COD CANAL.

1602 Cape Cod discovered by Bartholomew Gosnold.
1605 DeMonts, a Frenchman, cruised around Cape. Called it "Malebarre."
1620 Arrival of the Pilgrims.
1623 Miles Standish inspecting coast south of Plymouth.
1627 Standish trading at Manomet with Dutch from New Amsterdam. Boats within three miles of each other. Small canal talked of.
1676 Samuel Sewell examined canal route for "a passage from the South Sea to the North."
1697 General Court Massachusetts appoints a committee to view the place for a passage from Barnstable Bay to Manomet Bay.
1717 Captain Cyprian Southack describes a free water connection across the Cape. Colony of Massachusetts and many other individual examinations for shallow canal until
1776 General Washington ordered Thomas Machin to survey the route which would give greater security to navigation and *against the enemy*." Also James Bowdoin and William Sever appointed a committee to investigate subject.

1780 General Henry Knox using influence to have Government build canal.
1791 John Hills and James Winthrop make survey.
1798 James Sullivan and others petition legislature to build.
1803 Committee appointed by legislature to interest people in project.
1808 Gallatin, Secretary of Treasury, advocated it as "*useful in time of war.*"
1812 Isthmus used by small boats to avoid capture by *the enemy.*
1818 Israel Thorndike and Thomas H. Perkins employed Loammi Baldwin to survey route. Five thousand ships rounding the Cape. U. S. Senate made survey for a canal sufficient to admit *vessels of war.*
1824 U. S., under resolution of Senator Lloyd, ordered Major Perault to make survey. Varous army officers reported route practicable.
1825 Further surveys made under Government orders until 1827.
1828 Government Board reports on canal 36 feet wide, 8 feet deep, to cost $669,522.
1844 Lieutenant G. S. Blake, U. S. N., survey.
1860 Renewed interest by Massachusetts and Government. Surveys by Brigadier General J. G. Totten, Professor A. D. Bache, Superintendent Coast Survey, Commander C. H. Davis, U. S. N., and Lieutenant J. Wilkinson, U. S. N. Report on 18-foot canal to cost $10,000,000.
1870 Charter to Cape Cod Ship Canal Company to Alpheus A. Hardy and others. Major General J. G. Foster, U. S. A., examining route under orders. Reports on practicability of 23-foot canal *without locks.*
1870 Senator Chandler introduces Bill in Congress to pay for breakwaters.
1880 Charter granted to Cape Cod Canal Company, Henry M. Whitney and others interested. Some work done, but project abandoned.
1882 General G. K. Warren, U. S. A. Engineers, reports to Government on canal and cost of approaches.
1883 Cape Cod Ship Canal Company authorized; William Seward, Samuel Fessenden and others interested. Funds advanced by Quincy A. Shaw. Company made contract with F. A. Lockwood. Excavated over one million yards and failed. C. M. Thompson, Engineer.
1890 Property assigned to Colonel Thomas L. Livermore by F. A. Lockwood.
1891 Charter granted to E. L. Corthell and others in the name of Massachusetts Maritime Canal Company. This charter lapsed.
1898 DeWitt C. Flanagan, of New York, became interested.
1899 June 1, charter to the Boston, Cape Cod and New York Canal Company, under which and its amendments present Company acting.
During the three successive years Mr. Flanagan, General C. C. Dodge and associates advocating canal. Reports from the Engineers A. L. Rives and E. L. Corthell.
1904 Mr. August Belmont and others became interested.
1906 Voluminous report under instructions from William Barclay Parsons, C. E., from C. S. Sims to F. D. Underwood.
1907 May 8, approval by Joint Board of Railroad and Harbor and Land Commissioners of the present location of canal.
1909 June 22, work begun officially by Mr. August Belmont, at Bournedale.

TEST OF AN EXTERNAL-JOINT FILM OIL HEATER AT THE ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

The object of the test was to determine the rate of heat transmission in B.T.U. per square foot of heating surface per hour per degree F. temperature difference between the steam and the oil for various steam pressures and rates of flow of oil. The resistance to flow of oil through the heater and the amount of oil heated per pound of steam were also found.

The heater, as shown in Fig. 1, consists of a cylindrical composition casting containing two spirally-corrugated copper tubes, one of which screws into the other. The oil passes upwards through the heater as a film between the two tubes, the steam circulating downwards around the outer tube and within the inner tube. The steam joints and oil joints are so arranged that any failure will cause the steam or the oil, as the case may be, to leak into the atmosphere and thus be noticed, with the exception of the permanent connections between the inner corrugated tube and the castings at its ends. A defective tube might also cause a leak which would not be noticed immediately.

The following dimensions were obtained by measurement after the completion of the test:

Weight, empty, pounds	507½
Length over all, feet and inches	6–5
Width over external steam connections, inches	24
External diameter, inches	11½
Outer tube:	
Inside diameter, inches	4.03
Depth of corrugations, inch	0.62
Pitch of corrugations, inches	2
Thickness (assumed), B.W.G.	9
Length between headers, inches	43½

Inner tube:	
Outside diameter, inches	5.04
Depth of corrugations, inch	0.68
Thickness, inch	0.11
Length between heads, inches	54
Steam connections, inch	1
Oil connections, inches	1½
Distance between center lines of oil connections, inches	41½

The following values have been calculated from the above dimensions:

Outer surface of outer tube, square feet	5.58
Inner surface of inner tube, square feet	6.13
Total heating surface, square feet	11.71
Maximum inside diameter of outer tube, inches	5.27
Maximum outside diameter of inner tube, inches	5.04
Clearance, inch	0.115
Sectional area of oil film, square inches	1.86
Minimum inside diameter of outer tube, inches	4.03
Minimum outside diameter of inner tube, inches	3.68
Clearance, inch	0.175
Sectional area of oil film, square inches	2.12
Average sectional area of oil film, square inches	1.99

GENERAL ARRANGEMENT FOR TEST.

The general arrangement of the apparatus for the test is shown in Figs. 2 and 3. Provision was made to use the same oil over and over again by cooling it in the condenser *C*.

The cool oil enters the heater *H* at *a*, and when warmed leaves it at *b*. The warm oil enters the condenser at *c* where a longitudinal pipe is connected having a number of small holes to spray the oil over the condenser tubes. The cool oil flows from the condenser through the pipes and valves at *d* into the weighing tanks W_1, W_2 and W_3. In order to make more effective use of the cooling surface in the condenser with high rates of oil flow, the loop *L* was connected to the horizontal outlet pipe. When the valve in the horizontal outlet pipe is closed and the valve in the loop opened, the oil rises in the condenser to a slightly higher level than the top of the loop, thus securing the full effect of the cooling surface immersed

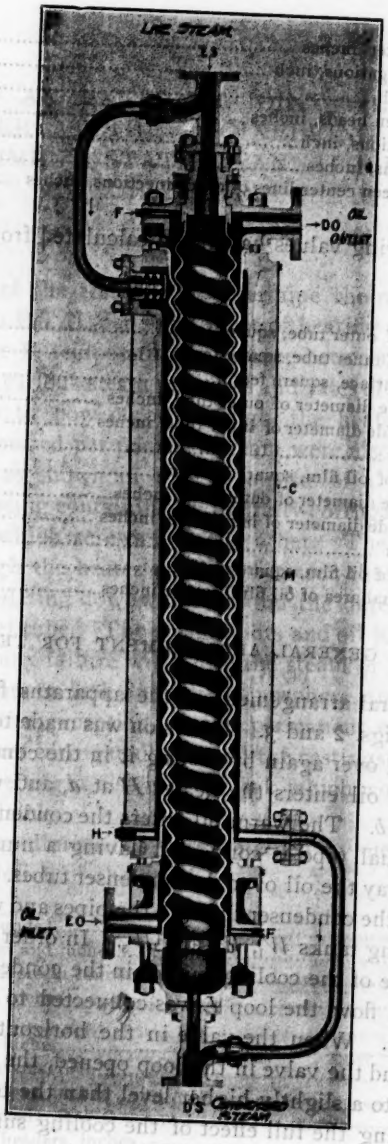


Fig. 1.

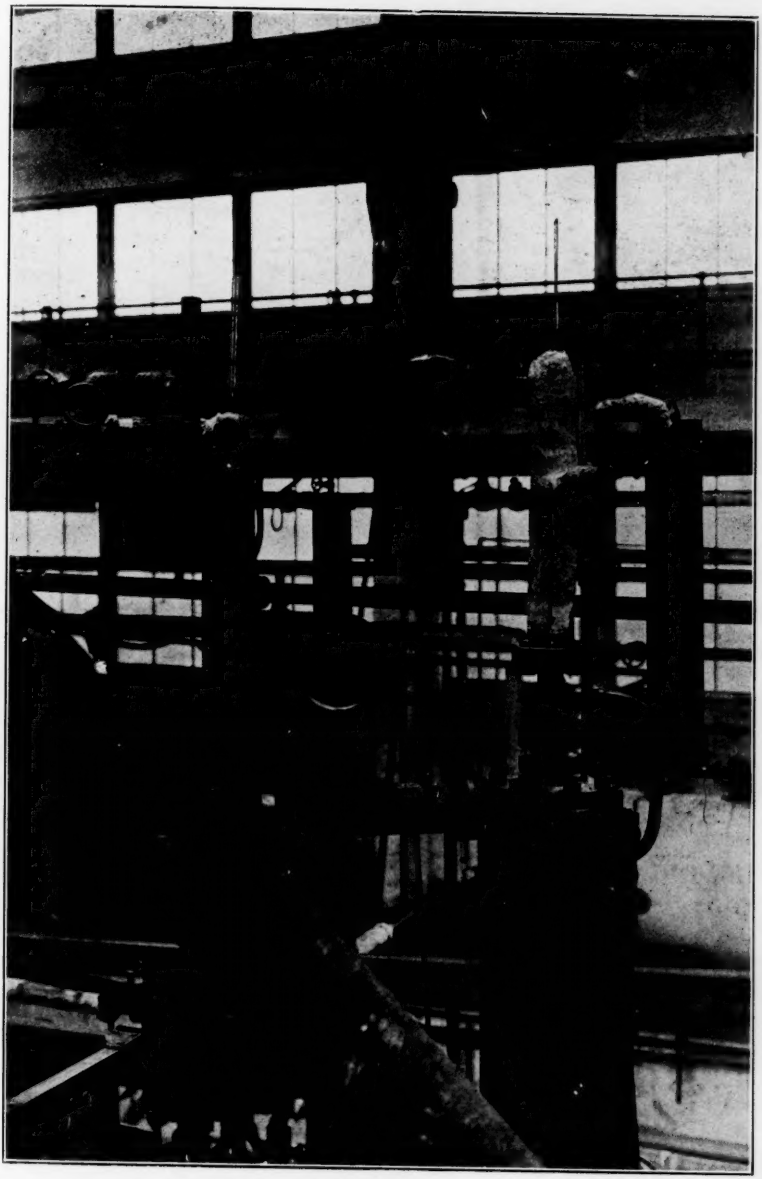


FIG. 2.

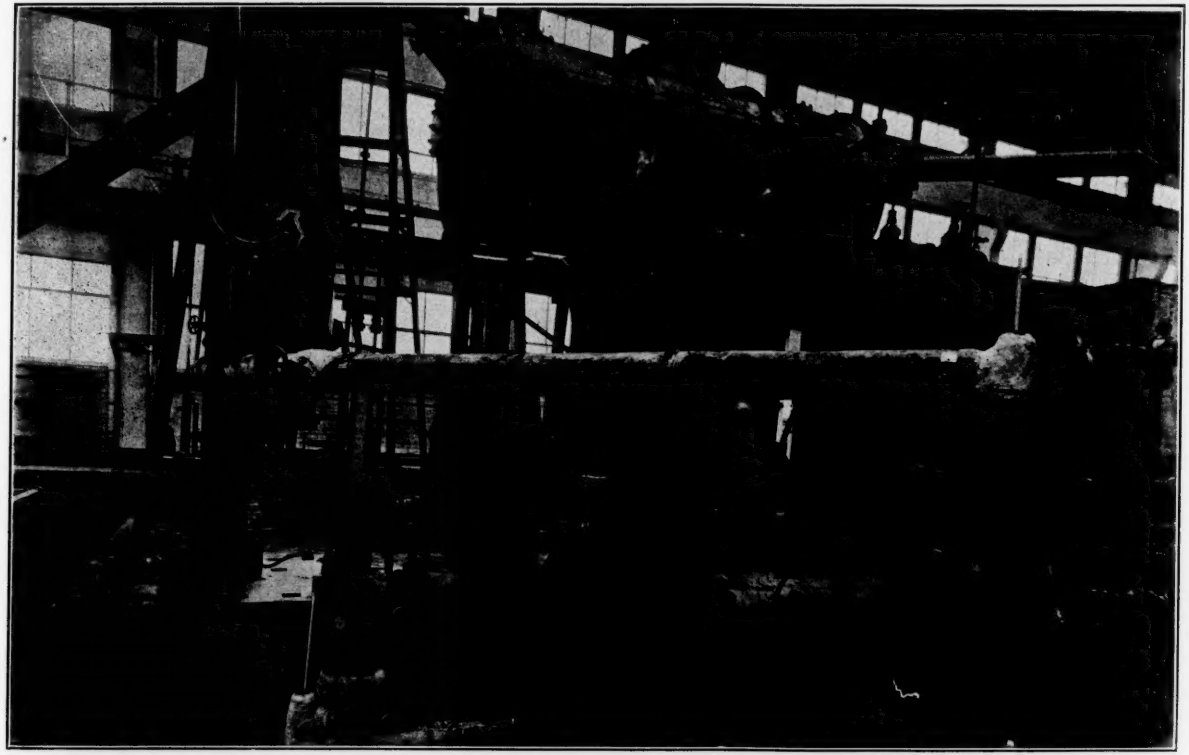


FIG. 3.

in the oil. From the weighing tanks the oil flows through the pipes at *e* into the storage tank *T*. An oil-circulating pump draws from the tank *T* and discharges at *f*, on which pipe an air chamber is mounted. The air chamber was filled with compressed air to secure the full benefit of its entire volume.

The steam enters the heater at *m* after having its moisture removed by a separator and its pressure reduced by the pressure-regulating valve *R*. The condensed steam leaves at *n*, dropping into the condensed-steam receiver *Q*. The water from *Q* passes through the pipe *r* and a valve to a surface condenser or cooling coil not shown. This condenser or cooling coil simply cools the condensed steam in order that no loss will occur by evaporation when this water is run into weighing tanks. The pet cock *P* on the condensed-steam receiver *Q* was provided to allow the air and other gases (brought over with the steam) to escape instead of filling up the heater. The fuel oil used in the test was purchased from The Texas Company, and showed by test the following characteristics: Specific gravity, 0.908 at 67 degrees F. and 0.866 at 160 degrees F.; specific heat, 0.476 between 212 degrees F. and 70 degrees F.; flash point, 226 degrees F. in open cup and 200 degrees F. in closed cup; fire point, 248 degrees F. in open cup.

CONDUCT OF TEST.

One series of runs was made during February, but these were discarded when discrepancies appeared in the heat balance, attributed to a leak in the condensed-steam line, to the location of the oil-throttling valve V_3 too near the oil outlet thermometer T_3 (the heat generated at V_3 being conducted to the thermometer T_3), and to the use of an inaccurate value of the specific heat of oil. When the second series of runs was made, any condensed steam leak was carefully avoided and the oil throttling valve V_3 was moved some distance beyond the oil outlet thermometer T_3. An accurate determination of the specific heat was made at the Bureau of Standards, Washing-

ton, D. C. Due to these precautions the second series of runs was more consistent than the first series.

Eighteen runs were made, each of one-hour duration, with approximately 4,000, 6,000 and 9,000 pounds of oil per hour, and steam at 10, 50, 100, 150, 200, and 250 pounds per square inch gage pressure. Three runs were made with steam only to determine the heat lost by radiation; and three of short duration with oil only to find the pressure drop through the heater with cold oil.

The rate of oil flow was measured by weighing the oil in the three tanks W_1, W_2 and W_3. Three tanks were found necessary to weigh 9,000 pounds of oil per hour, two tanks being emptied while one was filling. The flow was kept nearly constant during the run by adjusting the speed of the oil pump.

The pressure of the oil at the outlet, as indicated by the gage G_3, was kept at 150 pounds per square inch gage by adjusting valve V_3. The pressure drop through the heater was obtained by means of the differential gage D. The gage G_2 on the oil inlet was used when the pressure drop exceeded the maximum reading on the differential gage.

The temperature of the entering oil, as indicated by thermometer T_2, was kept near 100 degrees F., the flow of cooling water through condenser C being regulated during each run. This value is in accordance with the claims of the manufacturers based upon an entering oil temperature of 100 degrees F. Several runs, of short duration, however, were made to show the effect of a lower temperature upon the resistance to flow of oil through the heater. The temperature of the leaving oil was indicated by thermometer T_3.

The pressure of the entering steam was automatically reduced to the desired value by the pressure-regulating valve R. The temperature of the entering steam was obtained from thermometer T_1. The temperature and pressure in the calorimeter K were obtained from thermometer T_5 and manometer M, respectively.

The condensed steam was maintained at a constant level in

Run Number.	Date, March, 1912.	Duration of Test, Hours.	Barometer Corrected, Inches of Mercury.	Temperature of Room, Degrees, Fahr.	Oil.									Steam Condensed, Pounds per Hour.
					Oil Pumped Through Heater, Pounds per Hour.	Pressure at Outlet, Pounds per Square Inch. Gage.	Pressure at Inlet, Pounds per Square Inch, Gage.	Differential Gage Reading, Inches of Mercury.	Resistance Through Heater, Pounds per Square Inch.	Temperature at Inlet, Degrees, Fahr.	Temperature at Outlet, Degrees, Fahr.	Temperature Rise, Degrees, Fahr.	Heat Apparently Absorbed by Oil, B.T.U. per Hour.	
I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	XIII	XIV	XV
1	19	1	29.95	74.5	4794	154.1	——	18.1	9.6	105.1	169.6	64.5	138 000	147 1/4
2	19	1	29.92	71.3	4410	154.3	——	12.9	7.3	103.2	217.9	114.7	252 100	282 1/2
3	22	1	30.26	70.2	4261	154.8	——	7.8	4.9	95.3	254.2	158.9	398 800	464
4	22	1	30.26	71.1	4302	154.4	——	10.6	6.2	104.7	287.9	183.2	409 700	493 1/4
5	23	1	30.46	69.9	4547	154.4	——	11.2	6.5	103.2	300.5	197.3	458 700	565
6	23	1	30.41	70.3	4676	154.4	——	11.5	6.6	98.9	308.9	210.0	485 700	611 3/4
7	27	1	30.16	69.0	6651	156.7	——	30.1	15.1	109.7	174.3	64.6	182 700	189 3/4
8	27	1	30.00	74.3	6553	155.1	——	23.3	12.0	103.9	202.6	98.7	291 100	320 3/4
9	27	1	29.99	74.1	6712	155.9	——	26.3	13.4	105.0	235.0	130.0	421 400	489 1/2
10	27	1	29.97	74.7	6899	156.1	——	26.6	13.5	101.6	258.5	156.9	525 200	635 3/4
11	29	1	29.28	78.1	6974	155.3	——	24.3	12.5	100.2	282.1	181.9	608 900	765
12	29	1	29.30	77.5	7028	156.2	——	26.4	13.4	101.9	289.4	187.5	675 000	852
13	28	1	29.98	70.0	10222	155.3	187.6	——	33.7	104.1	163.9	59.8	249 000	255 1/2
14	28	1	29.95	72.6	10213	155.4	181.0	——	27.0	102.1	194.2	92.1	430 800	471 1/2
15	28	1	29.91	73.3	9850	154.6	180.8	——	27.6	104.9	230.1	125.2	589 000	686 1/2
16	28	1	29.88	71.1	10097	154.7	179.6	——	26.3	99.1	249.2	150.1	716 700	861 3/4
17	29	1	29.34	72.7	10083	155.2	179.6	——	25.8	99.1	260.9	161.8	786 900	981 1/4
18	30	1	30.16	77.0	10342	155.4	178.9	——	24.9	101.0	276.0	175.0	880 100	1112
19	30	1	30.13	70.1										5.4
20	14	1	30.15	67.7										9.5
21	15	1	29.52	66.1										16.0
22	30	1/6	30.12	72.4	4398	153.4	177.8	——	25.8	72.3	73.4	1.1		
23	30	1/6	30.12	73.0	6516	154.3	192.6	——	39.7	72.5	73.4	0.9		
24	30	1/6	30.13	72.6	10470	154.3	226.0	——	73.1	72.6	73.6	1.0		

STEAM.

STEAM CONDENSED, POUNDS PER HOUR.	PRESSURE AT INLET, POUNDS PER SQUARE INCH, GAUGE.	PRESSURE AT INLET, POUNDS PER SQUARE INCH, ABSOLUTE.	TEMPERATURE AT INLET, DEGREES, FAHR.	TEMPERATURE CORRESPONDING TO ABSOLUTE PRESSURE AT INLET, DEGREES, FAHR.	SUPERHEAT, DEGREES, FAHR.	TEMPERATURE IN CALORIMETER, DEGREES, FAHR.	PRESSURE IN CALORIMETER, INCHES OF MERCURY.	DRY STEAM, PERCENT.	TEMPERATURE OF CONDENSED STEAM AT OUTLET, DEGREES, FAHR.	EQUIVALENT DRY AND SATURATED STEAM, POUNDS PER HOUR.
XV	XVI	XVII	XVIII	XIX	XX	XXI	XXII	XXIII	XXIV	XXV
147 1/4	9.7	24.4	308.3	239.4	68.9	—	—	—	235.6	150.3
282 1/2	51.1	65.8	321.9	298.8	23.1	—	—	—	295.0	284.5
464	100.6	115.5	346.7	338.4	8.3	—	—	—	335.5	463.1
493 1/4	148.7	163.6	367.1	365.4	—	291.3	9.96	99.11	363.6	489.0
565	199.5	214.4	386.5	387.8	—	298.6	15.22	98.77	384.0	558.2
611 3/4	250.9	265.8	404.3	407.6	—	298.4	13.06	98.39	401.6	602.1
189 3/4	9.8	24.6	315.4	238.2	77.2	—	—	—	235.6	197.0
320 3/4	49.6	64.3	327.7	297.3	30.4	—	—	—	292.6	326.1
489 1/2	100.4	115.1	338.1	338.2	—	284.1	3.11	99.42	333.6	486.7
635 3/4	150.8	165.5	365.2	366.3	—	268.1	5.62	97.76	361.9	621.6
765	200.1	214.5	385.3	387.8	—	277.9	9.28	97.72	383.8	747.6
852	249.5	263.9	403.7	405.9	—	281.7	12.94	97.46	401.5	830.4
255 1/2	9.8	24.5	318.8	239.0	79.8	—	—	—	236.2	265.7
471 1/2	50.8	65.5	328.9	298.5	30.4	—	—	—	295.7	479.4
686 1/2	101.2	115.9	339.7	338.6	—	269.3	3.30	98.62	335.7	677.0
861 3/4	150.0	164.7	365.2	365.9	—	271.2	5.91	97.95	362.6	844.1
981 1/4	188.4	202.8	380.7	383.0	—	244.3	8.61	95.96	379.0	941.8
1112	248.9	263.7	403.6	405.8	—	269.8	12.55	96.78	400.5	1076.3
5.4	10.6	25.4	248.2	240.9	7.3	—	—	—	202.4	—
9.5	100.8	115.6	338.7	338.5	—	268.3	1.40	98.58	295.9	—
16.0	196.2	210.7	386.0	386.3	—	282.7	2.08	98.11	346.8	—

STEAM.

STEAM CONDENSED, POUNDS PER HOUR.	PRESSURE AT INLET, POUNDS PER SQUARE INCH, GAUGE.	PRESSURE AT INLET, POUNDS PER SQUARE INCH, ABSOLUTE.	TEMPERATURE AT INLET, DEGREES, FAHR.	TEMPERATURE CORRESPONDING TO ABSOLUTE PRESSURE AT INLET, DEGREES, FAHR.	SUPERHEAT, DEGREES, FAHR.	TEMPERATURE IN CALORIMETER, DEGREES, FAHR.	PRESSURE IN CALORIMETER, INCHES OF MERCURY.	DRY STEAM, PERCENT.	TEMPERATURE OF CONDENSED STEAM AT OUTLET, DEGREES, FAHR.	EQUIVALENT DRY AND SATURATED STEAM, POUNDS PER HOUR.	HEAT SUPPLIED BY STEAM, B.T.U. PER HOUR.
XY	XYI	XYII	XYIII	XIX	XX	XXI	XXII	XXIII	XXIV	XXV	XXVI
147 1/4	9.7	24.4	308.3	239.4	68.9	——	——	——	235.6	150.3	143 700
282 1/2	51.1	65.8	321.9	298.8	23.1	——	——	——	295.0	284.5	260 100
464	100.6	115.5	346.7	338.4	8.3	——	——	——	335.5	463.1	409200
493 1/4	148.7	163.6	367.1	365.4	——	291.3	9.96	99.11	363.6	489.0	420 100
565	199.5	214.4	386.5	387.8	——	298.6	15.22	98.77	384.0	558.2	469 900
611 3/4	250.9	265.8	404.3	407.6	——	298.4	13.06	98.39	401.6	602.1	497 600
189 3/4	9.8	24.6	315.4	238.2	77.2	——	——	——	235.6	197.0	188400
320 3/4	49.6	64.3	327.7	297.3	30.4	——	——	——	292.6	326.1	298800
489 1/2	100.4	115.1	338.1	338.2	——	284.1	3.11	99.42	333.6	486.7	430 500
635 3/4	150.8	165.5	365.2	366.3	——	268.1	5.62	97.76	361.9	621.6	535300
765	200.1	214.5	385.3	387.8	——	277.9	9.28	97.72	383.8	747.6	629600
852	249.5	263.9	403.7	405.9	——	281.7	12.94	97.46	401.5	830.4	686400
255 1/2	9.8	24.5	318.8	239.0	79.8	——	——	——	236.2	265.7	253 900
471 1/2	50.8	65.5	328.9	298.5	30.4	——	——	——	295.7	479.4	437900
686 1/2	101.2	115.9	339.7	338.6	——	269.3	3.30	98.62	335.7	677.0	597 500
861 3/4	150.0	164.7	365.2	365.9	——	271.2	5.91	97.95	362.6	844.1	726300
981 1/4	188.4	202.8	380.7	383.0	——	244.3	8.61	95.96	379.0	941.8	797 100
1112	248.9	263.7	403.6	405.8	——	269.8	12.55	96.78	400.5	1076.3	890 900
5.4	10.6	25.4	248.2	240.9	7.3	——	——	——	202.4	——	5 370
9.5	100.8	115.6	338.7	338.5	——	268.3	1.40	98.58	295.9	——	8660
16.0	196.2	210.7	386.0	386.3	——	282.7	2.08	98.11	346.8	——	13840

B.T.U. per Hour.	Heat Lost by Radiation, B.T.U. per Hour.	Heat Gained by Friction, B.T.U. per Hour.	Apparent Specific Heat of the Oil.	Velocity of Oil through Tube, Feet per Minute.	Heat Transfer per Square Foot of Heating Surface per Degree of Maximum Temperature Difference (Col. XII - Col. XI) B.T.U. per Hour.	Mean Temperature Difference Assuming Heat Transfer Proportional to Temperature Difference Degrees, Fahr.	Heat Transfer per Square Foot of Heating Surface per Degree of Mean Temperature Difference, B.T.U. per Hour	Dry Steam Condensed per Square Foot of Heating Surface, Pounds per Hour.	Oil Heated per Pound of Dry Steam, Pounds.
XYI	XXYII	XXYIII	XXIX	XXX	XXXI	XXXII	XXXIII	XXXIY	XXXY
3 700	5860	152	0.446	80.9	87.7	98.6	119.5	12.84	33.09
60 100	8080	106	0.498	74.4	110.1	129.8	165 9	24.30	15.50
09200	9530	69	0.589	71.9	140.1	149.9	227.3	39.55	9.20
20 100	10460	88	0.520	72.6	134.2	151.0	231.7	41 76	8.80
69 900	11300	98	0.511	76.7	137.6	167.0	234.6	47.67	8.15
97 600	11980	102	0.495	78.9	134.4	184.1	225 3	51 42	7 77
88400	6010	331	0.425	112.2	121.4	92.5	168.7	16.82	33 76
98 800	7920	259	0.450	110.5	128.6	138.3	179.8	27.85	20.10
30 500	9380	297	0.483	113.2	154.3	159.5	225 6	41 56	13.79
35 300	10360	307	0.485	116.4	169.5	174.7	256.8	53.08	11.10
29 600	11000	288	0.493	117.6	180.8	181.8	286.0	63 84	9 33
86400	11670	311	0.512	118.5	189.6	195.5	294.9	70.91	8.46
53 900	6000	1136	0.407	172.4	157.7	102.1	208.3	22.69	38.47
37 900	8030	909	0.458	172.2	187.3	145.5	252.8	40.94	21 30
47 500	9430	896	0.478	166.1	215.2	163.1	308.4	57.81	14.55
26 300	10470	876	0.473	170.3	229.4	181.4	337.4	72.08	11 96
97 100	11030	858	0.482	170.1	236.7	191.8	350.4	80.43	10.71
90 900	11680	849	0.486	174.4	246.6	205.0	366.6	91.91	9.61
5 370	Average Heat Radiated per Hour per Degree of								
8660	Temperature Difference between Steam and								
13 840	Room, 35.53 B.T.U.								

TABLE I.

the condensed-steam receiver Q by regulating the valve V on the outlet pipe r. Its temperature was obtained from thermometer T_4. During each run the pet cock P was adjusted until a small amount of vapor issued from it.

All readings of thermometers and gages were taken at five-minute intervals during runs 1 to 21, and at two-minute intervals during runs 22 to 24, with the exception of the barometer, which was read at the beginning and end of each run. The condenser was tested and found tight. All scales, gages and thermometers were carefully calibrated before and after the test and corrections applied.

OBSERVATIONS AND DEDUCED RESULTS.

The observations and deduced results are tabulated in Table I. Runs 1 to 18 inclusive show the operation of the heater when 4,000, 6,000 and 9,000 pounds of oil per hour are heated by steam having pressures of 10, 50, 100, 150, 200 and 250 pounds per square inch gage respectively. Runs 19 to 21 inclusive were made with steam only to determine the radiation from the heater. Runs 22 to 24 inclusive were made with oil only to find the resistance through the heater with cool oil.

The source of each column in Table I is given below, Marks & Davis Steam Tables being used:

Column.

I. For reference.

II. From data taken during test.

III. From data taken during test.

IV. From data taken at beginning and end of each run corrected by Smithsonian Meteorological Tables.

V. From data taken during each run.

VI. From weighings made during each run.

VII. From data taken during each run.

VIII. From data taken during each run.

IX. From data taken during each run. During runs 13 to 18 and 22 to 24 the resistance was beyond the range of the differential gage.

X. Column IX multiplied by 0.458 and then increased by 1.35 pounds per square inch. In runs 13 to 18 and 22 to 24, the resistance is column VIII minus column VII plus 1.35 pounds per square inch. The factor 0.458 is obtained from $(13.58 - 0.90) \times 0.0361$, where 13.58 is the specific gravity of mercury in one leg of the gage, 0.90

Column.

is the specific gravity of the oil in the other leg of the gage, and 0.0361 is the number of pounds per square inch equivalent to one inch of water. The value 1.35 pounds per square inch is the pressure equivalent to the head of oil, 41½ inches between the oil inlet and outlet pipes; for, using a specific gravity of 0.9 for the oil, we have $41\frac{1}{2} \times 0.90 \times 0.0361 = 1.348$ pounds per square inch.

XI. From data taken during each run.

XII. From data taken during each run.

XIII. Column XII minus column XI.

XIV. Column XXVI minus column XXVII plus column XXVIII.

XV. From weighings made during each run.

XVI. From data taken during each run.

XVII. Column XVI plus pressure corresponding to barometer reading in column IV.

XVIII. From data taken during each run.

XIX. From steam tables corresponding to pressure in column XVII.

XX. Column XVIII minus column XIX.

XXI. From data taken during each run.

XXII. From data taken during each run.

XXIII. From formula $\frac{(H + k(t - t_p)) - q}{r} \times 100$, where q is the heat of the liquid and r is the latent heat corresponing to pressure in column XVII, H is the total heat and t_p is the temerature corresponding to the pressure in column XXII, t is the temperature in column XXI, and k is the specific heat of the superheated steam in the calorimeter.

XXIV. From data taken during each run.

XXV. From formula $W \times \frac{H}{K}$, where W is the number of pounds of steam condensed per hour in column XV, K is the heat required to produce dry and saturated steam from the temperature in column XXIV, and at the pressure in column XVII, and H is the heat required to produce the actual steam from water at the temperature in the column XXIV. In the case of wet steam, H is the sum of the latent heat of the percentage of dry steam given in column XXIII and the heat of the liquid at the pressure in column XVII decreased by the heat of the liquid, corresponding to temperature in column XXIV. In the case of superheated steam, H is the total heat of steam at pressure in column XVII and superheated in column XX decreased by the heat in the liquid corresponding to the temperature in column XXIV.

XXVI. Column XV multiplied by the value of H determined as explained above.

XXVII. From formula $35.53\ (t_s - t_r)$, in which t_s is the the temperature corresponding to the steam pressure, column XIX, t_r is the temperature of the room, column V, and the factor 35.53 *B.T.U.* is the heat radiated per hour per degree difference in steam and room temperature from runs 19 to 21.

Column.

XXVIII. From formula $H \times 778 = \frac{F \times 2.309}{0.9} \times W$, in which H is the heat generated by friction, F is the resistance, column X, and W is the oil flow per hour, column VI. The constant 2.309 is the head of water equivalent to one pound per square inch and 0.9 is the specific gravity of the oil.

XXIX. Column XIV divided by the product of columns VI and XIII.

XXX. From formula $\frac{W}{60\,A\,D}$; where W is the number of pounds of oil per hour, column VI, A is the average area by measurement between the two corrugated tubes, $A = \frac{1.99}{144} = 0.0138$ square foot, and D is the density of the oil, $D = 62.36 \times 0.866 = 54.0$ pounds per cubic foot.

XXXI. The difference between columns XXVI and XXVII divided by the number of square feet of heating surface by measurement on the corrugated tubes, 11.71 square feet, and by the difference of the temperatures in columns XIX and XI.

XXXII. From formula $D_m = \frac{t_a - t_i}{\log \frac{t_s - t_i}{t_s - t_o}}$; where t_s is the steam temperature, column XIX, t_i is the oil inlet temperature, column XI, and t_i is the oil outlet temperature, column XII.

XXXIII. The difference between columns XXVI and XXVII divided the number of square feet of heating surface by measurement on the corrugated tubes, 11.71 square feet, and by the mean temperature difference, column XXXII.

XXXIV. Column XXV divided by 11.71 square feet heating surface.

XXXV. Column VI divided by column XXV.

EXAMINATION OF APPARATUS TESTED.

At the conclusion of the first series of runs the heater was taken down and dismantled. The surfaces of the corrugated tubes were found coated with a layer of oil. The heater was reassembled and filled with a solution of water containing about one-half pound of lye. This was allowed to boil for one afternoon, low-pressure steam being admitted to the heater in the usual manner. The heater was again dismantled the following morning. The oily matter had been removed, but the tubes were stained black, and on the lowest corrugations the black deposit had an appreciable thickness. Some of the oil was boiled in a test tube until a similar deposit occurred on the surface of the test tube. Boiling lye, acid or

gasoline had no effect upon the deposit. The deposit is evidently asphalt and may best be removed by mechanical means.

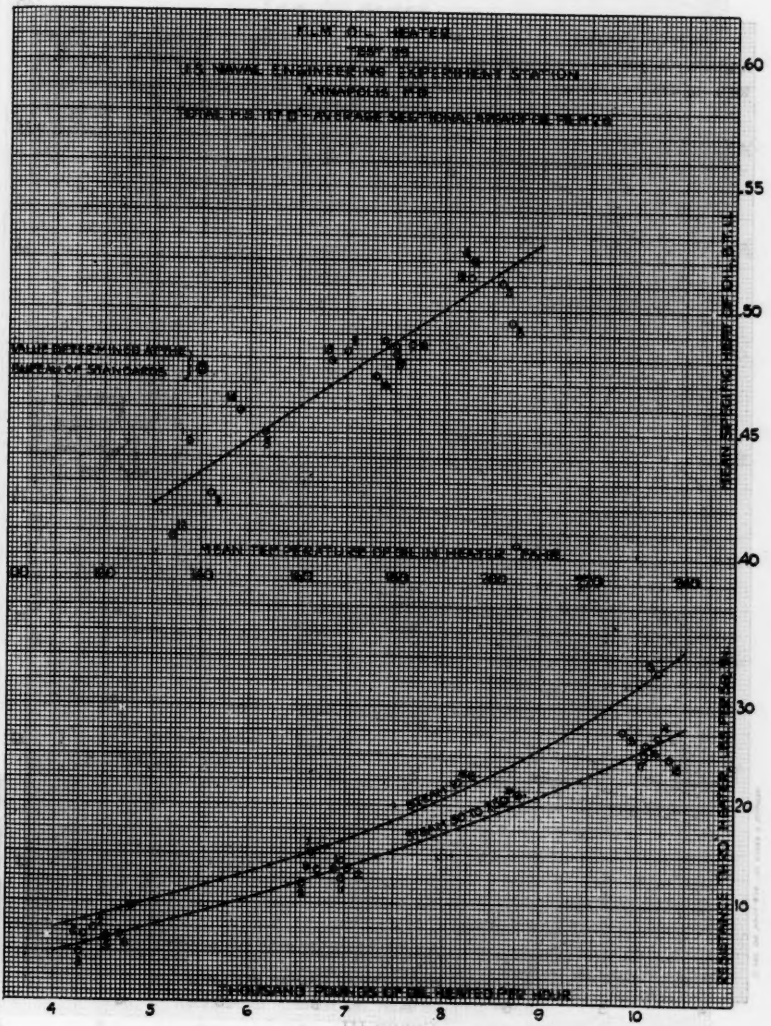
A longitudinal crack about ⅜ inch long was discovered at the top of one of the corrugations of the inner tube one-third its length from the top. This crack had been partly filled with soft spelter. The tube was found perfectly tight, however, when subjected to 200 pounds per square inch water pressure. The outer tube was likewise found to be without a leak.

Before reassembling for the second series of runs the oil surfaces of the tubes were carefully cleaned with emery cloth. The outer surface of the inner tube was cleaned by hand, but to clean the inner surface of the outer tube it was necessary to make a device which could be worked from one end of the tube. When the second series of runs was completed the heater was again dismantled, the oil space having been cleaned by blowing steam through it. The same thin, black deposit covered the tubes, and several pieces of waste were found in the oil space.

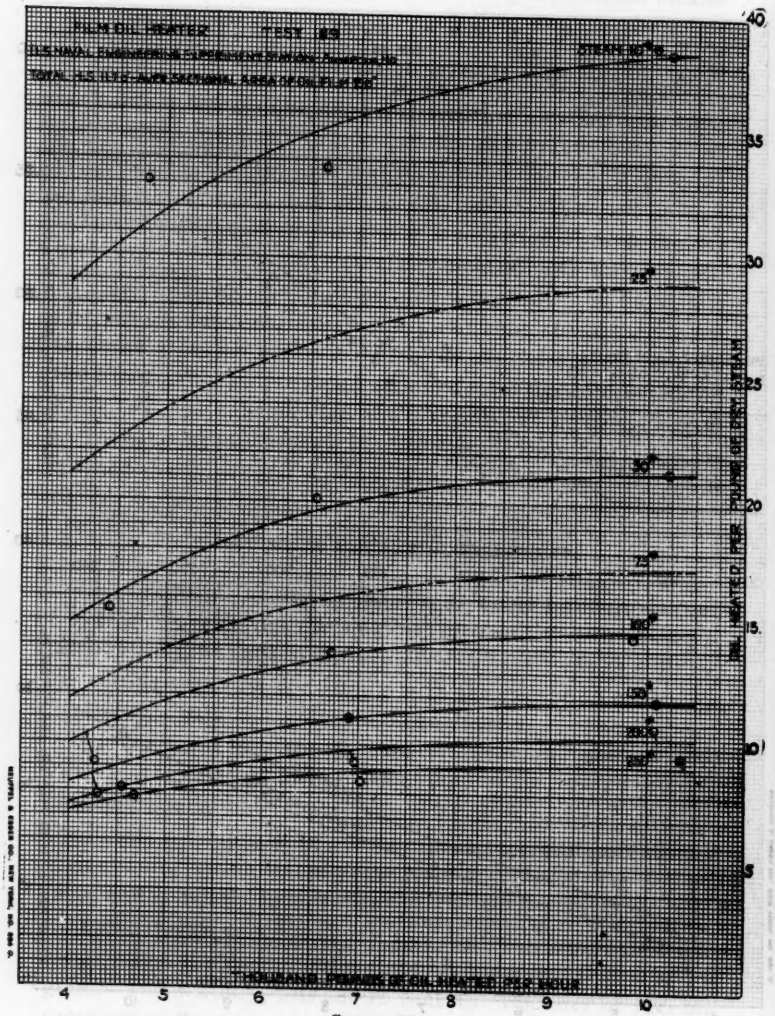
SUMMARY AND CONCLUSIONS.

The results of the test are shown in Table I and in Curves I to IV. Considering the number of quantities to be regulated during the runs and the interrelation of these quantities, the results are fairly consistent, with the exception of Run 3. The results may be summarized as follows:

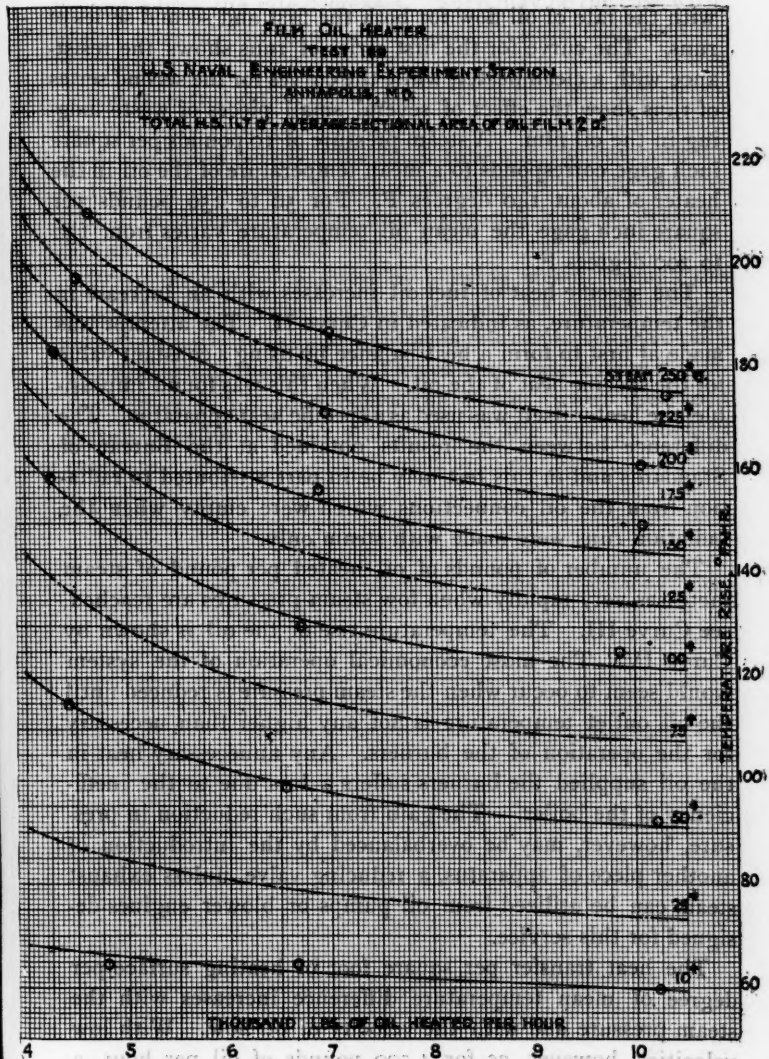
The resistance to the flow of oil through the heater is directly proportional to the rate of flow, is less than 5 pounds per square inch when 6,000 pounds of oil is pumped through the heater per hour, with an entering temperature of 100 degrees F. and with a steam pressure of 200 pounds per square inch, and is nearly independent of the oil temperature when its mean value exceeds about 150 degrees F., but does increase when the mean temperature becomes less than 150 degrees. Incidentally, this shows that the viscosity of fuel oil decreases with a rise in temperature to about 150 degrees F., beyond which there is little change. This conclusion is



CURVES I AND II.


FILM OIL HEATER
TEST #3
STEAM 80 lbs
25°
30°
75°
100°
150°
200°
250°
40
35
30
25
20
15
10
5
4
5
6
7
8
9
10

CURVE III.


FILM OIL HEATER.
TEST 188.
U.S. NAVAL ENGINEERING EXPERIMENT STATION.
ANNAPOLIS, MD.
TOTAL H.S. 1.7 SQ. FT. AVERAGE SECTIONAL AREA OF OIL FILM .2 SQ. FT.
STEAM 250°
225°
200°
175°
150°
125°
100°
75°
50°
25°
10°
TEMPERATURE RISE °FAHR.
220
200
180
160
140
120
100
80
60
THOUSAND LBS. OF OIL HEATED PER HOUR.
4
5
6
7
8
9
10

CURVE IV.

drawn from Curve I, which shows the variation of resistance with the rate of oil flow for both the first and second series of runs. The curve for the first series of runs shows the resistance with a clean heater, while the curves for the second series of runs show the effect of several small pieces of waste lodging in the oil film. A steam pressure of 10 pounds per square inch gage corresponds to a mean temperature of the oil in the heater of about 140 degrees F. For 50 to 250 pounds per square inch gage the mean oil temperature varies from 150 to 200 degrees F.

The specific heat of fuel oil increases with an increase in the temperature, as indicated in Curve II. The attempt was made at first to form a heat balance by using a constant value for the specific heat of the oil. When this was found impossible, the heater itself was used as a specific heat apparatus. The difference between the results obtained at the Bureau of Standards and in this test may be in part attributed to radiation from the oil connections which were empty when the radiation runs were made with steam only.

The number of pounds of oil heated per pound of steam increases very rapidly when low steam pressures are reached, see Curve III. The temperature rise of the oil is shown by Curve IV. The most economical operation of the system would seem to occur when the steam pressure is reduced until the oil outlet temperature is not any higher than necessary for the operation of the burners. Any unnecessary heat in the oil supplied the burners suffers a loss due to the inefficiency of the boiler. The gain from such reduction in pressure, however, may be overbalanced by the introduction of another piece of apparatus, a reducing valve, unless exhaust steam can be utilized from oil pumps or blower engines designed for this service.

The heat transfer per square foot of heating surface per degree of mean temperature difference increases with the steam pressure and with the velocity of oil flow. With low velocities, however, as for 4,500 pounds of oil per hour, a maximum rate of heat transfer is reached at some definite

steam pressure, about 200 pounds per square inch in this case, beyond which the rate decreases. Such may be the case with higher rates of flow at a higher steam pressure than used in this test.

The construction of the heater is such that oil and steam can mix only through the failure of the metal of either tube or of the joints between the inner tube and the composition castings at its ends. A failure at any joint which must be broken in dismantling the heater causes a leak of steam or oil into the atmosphere.

While oil can be removed by blowing steam through the plugged holes provided for this purpose, the asphalt deposit occurring with Texas and California oils can be removed only by dismantling the heater. This can be quickly done if sufficient space is provided below the heater (when used in a vertical position) for the withdrawal of the inner corrugated tubes.

THE *MONITOR* AND THE *MERRIMAC*.

By WALTER M. MCFARLAND, Member.*

Fifty years ago, on the 7th of March, the navies of the world were still rejoicing in the strength of their "wooden walls," even as Themistocles and the Athenians had done at the battle of Salamis about 2,400 years before. In many ways naval tactics had changed but little from that early day, and the physical prowess of the hardy seaman was still a vital factor in battle. Gunpowder had increased the distance at which an engagement began, and a limited use of steam, which was tolerated as a disagreeable necessity, gave increased mobility and partial independence of the wind. The full-rigged line-of-battle ship and the frigate, with her cloud of snowy canvas, were still the pride of the fighting men in all navies. The vessels and the tactics which had sufficed for Blake and Nelson, for Van Tromp and DeRuyter and for John Paul Jones and Decatur were good enough for their successors. There were no great fleets gathered in hostile array to settle questions of naval architecture at the same time that they fought for a national policy, and the world had little thought of the momentous event which was so near. And yet, the happenings of the next day, March 8th, ended all reliance upon "wooden walls," showed that the day of personal physical prowess was over, and that the only dependable war vessel was the fighting machine—the creature of steam and iron. The duel of the next day, March 9th, whose fiftieth anniversary we are celebrating, emphasized more fully the lessons of the previous day, but lessons are for

* Read at the banquet of the American Society of Naval Engineers, Washington, D. C., on March 9, 1912, the 50th anniversary of the battle between the *Monitor* and *Merrimac*.

those with eyes to see and ears to hear. It ought to have been plain that the fighting machine, which had proved that it was the survival of the fittest, should be officered by engineers and manned by mechanics, but it took us nearly forty years to learn the lesson. The change in war-ship design could not, however, be made all at once, even physically, and naval prejudices were also a cause of delay for years, but the new era had begun. It is most appropriate that the semi-centennial of this first combat between engineers' ships should be commemorated by this Society of Naval Engineers, especially now, when every naval officer is a fighting engineer and every seaman a fighting mechanic.

The circumstances preceding and attending the fight between the *Monitor* and the *Merrimac* are so important historically, and the effect of the two days' fight at Hampton Roads has been so tremendous upon naval design that what, under other circumstances, would have been merely an indecisive contest has taken rank as one of the great naval battles of history. It is hardly necessary to remind this audience, which is so well posted technically, that the *Monitor* and the *Merrimac* were not the first armored vessels. Indeed, a projected armor-clad, the Stevens Battery, had been in process of construction for many years. Nor was it the first case when armor-clads were under fire, because three French vessels took part in an engagement against the Kinburn forts in 1855, during the Crimean War. Both England and France had a number of armored vessels in commission at this time, but there had been no strictly naval action in which the armor-clad was pitted against the wooden vessel until March 8th, 1862.

It is so natural that all the glory in any contest should go to the victor that we hear much more about the details of the *Monitor* than of the *Merrimac*, and it seems to me that we are apt to give too little credit to the technical officers of the Confederate Navy who turned the hulk of the frigate *Merrimac* into the armor-clad ram which was called the *Virginia.* She was certainly not a very beautiful piece of naval architecture, and it is questionable whether she would have been a very

good sea boat; but the fact remains, that for one day she held the position of the most dreaded war vessel afloat. You are all familiar with her appearance, and I wish only to remark that her armor was composed of two layers of iron plates, each two inches thick, eight inches wide and ten feet long, and not of railroad rails, as is so often stated. The original *Merrimac* was a sailing frigate with auxiliary steam power, and, as the changes were all made in her upper works, she obviously was slow. Her pre-eminent features were the almost impenetrable armor surrounding a minimum gun platform, the entire absence of masts and top hamper, and the ram.

It is hard to say with just what feelings the men on the wooden vessels of the Federal Navy awaited the approach of the *Merrimac*, but when we remember that there were present several sailing vessels without even auxiliary steam power, two of which, the *Congress* and the *Cumberland*, were destroyed by the ram, it would look as though she was not deemed to be invincible. I need not go into the details of the combat of March 8th, because you all know them so well, but it proved that against the best ordnance of the day the armor of the *Merrimac* was impenetrable, and that, having destroyed a part of it, the rest of the wooden fleet at Hampton Roads was absolutely at her mercy and convenience. History tells us of the consternation all through the North at the destruction wrought by this new type of naval vessel. It was actually believed by many that, after destroying the rest of the fleet in Hampton Roads, the *Merrimac* would proceed to the Northern cities and lay them under tribute. It must have caused many a bitter regret on the part of the responsible naval officials that they had not earlier appreciated the importance of armor, so as to have at least a few armor clads in the Federal Navy ready for service. The elation of the Confederates was correspondingly great, and my classmate, Captain Bennett, in his "History of the Steam Navy," allows himself to imagine the thoughts of Catesby Jones, who succeeded Buchanan in command of the *Merrimac*, as he may well have dreamed of himself as the arbiter of the war then

going on and a possible President of the Southern Confederacy.

The Federal authorities had not been entirely indifferent to the possible value of armored vessels, and a Board of distinguished naval officers had been appointed to consider the subject in 1861. One is struck by the fact that there was no naval constructor on this Board, but he is amazed to learn that the Board asked that one should be detailed to assist them and were denied. They say, "It appears that they are all so employed on important service that none could be assigned to this duty." Could anything show more thoroughly how little was the fact appreciated that the wooden age was about to give way to that of iron? The Board submitted a report which can certainly not be called enthusiastic as to the value of armored vessels, and indeed took the view that it would probably be impracticable to construct them for regular cruising service. It did, however, believe that as floating batteries of light draught for service in inland waters they might be useful, and recommended that some be built. Familiar as we are today with the enterprise of the technical bureaus of our Navy, it seems almost inconceivable that no plans for an armored vessel were prepared by the Bureau of Construction in 1861, although the "important service" (whatever it was) on which the constructors were all employed may account for it. The Act of Congress of August 3, 1861, provided for the examination by the Board of officers of the "plans and specifications which may be submitted," and does not limit them to those originating outside the Service. It is also astonishing that no individual naval officer had any plan. Seventeen propositions were submitted, all by civilians. It was in accordance with the recommendation of this Board that the *Monitor* was one of three armored vessels to be contracted for.

The whole story of the design and construction of the *Monitor* is so romantic and so interesting that, if time had permitted, the Committee would have been only too glad to ask the distinguished historian of Captain Ericcson to tell it

to us. It is a record of good work against many obstacles and, if told in detail, would show how great events some times hang upon matters which seemingly have no connection with them. The design of the *Monitor* was different from anything which had ever appeared as a fighting vessel, and naturally aroused the derision of those who find any variation from the commonplace amusing, and whose chief aim in life is to make a clever epigram. You all know how she was referred to as "a cheese box on a raft." However, Ericcson himself had all the temperamental enthusiasm of the genius, and both he and the country were singularly fortunate in the man who was detailed as Government Inspector, Chief Engineer Alban C. Stimers. He made himself thoroughly familiar with every detail of the ship and knew it so well that he could operate every valve and lever. She was pre-eminently an engineers' ship, because without steam and its skilful application she would have been not only helpless, but what the monitors were afterwards sometimes called, "an iron coffin for a crew."

It was known that the work of transforming the *Merrimac* into an armor-clad was being pushed rapidly by the Confederates, and in consequence work on the *Monitor* was also hastened as much as possible. She was sent to sea on her way to Hampton Roads before she had been thoroughly tested, but the demand for her services would not permit any shaking-down trials. It was a strenuous voyage, and, on several occasions it seemed as if the vessel would be lost. She certainly would never have survived but for the heroism of all on board, and especially of Stimers and Newton, the engineers. She arrived at Hampton Roads late in the afternoon of March 8, just in time to hear the last shots of the battle which marked the downfall of the wooden fleets of the world. Officers and men were worn out by over exertion and loss of sleep; but there was no time for rest; the exigency for which she was constructed was at hand and she must prove her value.

I wonder if we can really do full justice to the courage and

fortitude of the gallant crew of the *Monitor.* Knowing the actual result of the combat with the *Merrimac,* are we not prone to think that they must have known it in advance and gone into the action with merry hearts sure of victory? On the contrary, it seems to me they may have felt anything but confidence. They were only about one-fourth the size of their adversary, and had only two guns against ten. These were heavier, it is true, than any on the other ship, but nobody had ever had any experience as to the effect of projectiles against armor at close range. As already stated, everybody was exhausted. Indeed, Stimers and Newton had been nearly asphyxiated on the way down, owing to the failure of the ventilating blowers, and Newton was sick in bed. Neither officers nor men had had time to become familiar with the machine which they had to handle; such experience as they had gone through was rather calculated to shake their confidence. I feel, therefore, that we cannot admire too highly their steadfastness and bravery, and they deserve even more credit for going into the fight than they received for their victory.

To tell the story of the fight of March 9th as it deserves, one must be an orator, to which I can lay no claim. The circumstances would enthuse the dullest mind, and they have many a time inspired both poetry and oratory. The disparity in size undoubtedly suggested at once the combat between David and Goliath, and, indeed, the whole combat was picturesque in the highest degree. I want to emphasize, as I have already remarked, that this was the first fight between engineers' ships, by which I mean that both vessels were steamers with no other method of propulsion, that there was no thought of boarding, and that as an example of utilization of the powers of nature, which is the engineer's special function, these vessels marked an era in naval history. It was no longer the physical courage of personal combat and the brawny arm, but the higher moral courage of the cool scientist, knowing the value of the weapons at his command and possessed of the skill and ability to utilize them with the highest efficiency. Incidentally there was an illustration of

the great value of the method of naval organization and training to which we came nearly forty years afterwards in having every officer a trained engineer as well as a fighting man. As already mentioned, Chief Engineer Stimers had made himself absolutely familiar with every detail of the *Monitor*. He was not a part of her regular complement, but went as a volunteer to assist in her operation. After the heroic Worden had been wounded and the gallant young lieutenant, Greene, had left the turret and taken his place in the pilot house, as commanding officer, Stimers took command of the turret, where he had been during the entire engagement, to insure that it properly answered its function as a gun carriage.

It is most interesting to consider how this combat was a foreshadowing of the conditions of a naval action of today. As has been cleverly pointed out by Capt. Van Duzer, of our Navy, if the sides of the *Merrimac* were made vertical and a turret put at each end, we would have the battleship prior to the Dreadnought type, and this latter only adds a few more turrets. In other words, in the two combatants of 1862 we have the essential elements of the battleship of today—the great fighting machine—the creature of the engineer and dependent on him to operate it with complete efficiency.

From one point of view the combat was indecisive, in that neither of the armor clads was seriously injured or forced to surrender, but from another it was a great victory for the *Monitor*, because the *Merrimac* had come out expecting fully to destroy the Federal ships and was compelled to withdraw, having entirely failed in her mission. It was no wonder that the praises of the *Monitor*, of her designer Captain Ericcson, and of the officers and crew of the vessel were in every mouth. The people of the North felt that they had been delivered from the attacks of a terrible monster, and they could not do too much to show their gratitude and rejoicing. The victory led, as could be expected, to the construction of a large number of monitors, and for many years this was looked upon as the distinctively American type of armored vessel. Foreign navies built a few of them, but perhaps a certain amount of

prejudice and possibly a correct view of naval design led them to the conclusion that, for deep-water cruising, a vessel of much higher free board was necessary. I was much struck in looking over the article on shipbuilding in the latest edition of the Encyclopedia Britannica, where the subject of armored ships is treated, to notice an expression which, in view of all our knowledge, seems highly significant. After telling of the early work of building iron-clads in England, of which the *Warrior* was the first, in 1859, the paragraph begins: "It was in 1863 that the real reorganization of the British Navy occurred, under Sir Edward Reed." You will note that this was very soon after the *Monitor* and *Merrimac* fight.

There are many lessons to be drawn from this historic event, but time will not permit a consideration of all of them. I wish, however, to dwell just a little on one phase of the subject. There has been an unwarranted tendency in our country to trust to luck or Providence to get us out of trouble, relying on what we like to believe is a peculiarly American characteristic, ingenuity. In other words, it seems to be believed that we can let the world move on in military affairs, with constant improvements, waiting until the necessity arises before we attempt to meet it. We have not been guilty of this in recent years, as our present magnificent fleet shows, but there are constant evidences of this spirit in certain quarters and it it is by no means dead. Such optimists might be inclined to cite the case of the *Monitor* and the M*errimac* as conclusive proof that, when the occasion arises, we shall be able to meet it. I do not think a fair analysis of the case warrants any such conclusion. On the contrary, the facts are exactly the reverse. It is to be remembered that, if the Union Navy was not up to date, the Confederates had no navy at all. The *Merrimac* herself was the rehabilitation of a vessel which had been partly burned and sunk, yet the armor-clad hastily constructed out of this wreck proved invincible against the vessels of the Union Navy which were within twenty miles of where she was building and which awaited her. Have we the right

to be so very proud of the ingenuity which enabled us to conquer such a temporary construction, and this the only one of its kind? Suppose that France, which then had at least half a dozen armor-clads, had backed up the South, what could the little *Monitor* have done against so many?

No. In the words of the great Jefferson, "The price of liberty is eternal vigilance," and in military matters we can paraphrase this and say that "the price of independence and peace of mind is eternal preparedness." Battleships cannot be built in a day, nor even in a hundred days, as the *Monitor* was. It is a matter of years. The other great nations of the world are systematically keeping their fleets in a condition of the highest efficiency, and if the Great Republic of the West is to deserve and maintain her place among the nations, she must do the same. We are rightly urged to have a definite "naval program," and this is discussed as if it were a new idea. I confess that I was surprised myself to find recently that, almost five hundred years before our era, Themistocles, after the battle of Marathon, and realizing that the final question of Persian or Grecian supremacy would be fought out on the sea, prevailed upon the Athenians to establish a naval program by utilizing the output of the silver mines at Laurium to build each year a certain number of triremes, instead of distributing it among the citizens as had been at first intended. The fleet which resulted enabled the Athenians to win the great victory at Salamis.

It must strike you at once that there is a remarkable similarity between this ancient history and the condition which confronts our Navy today. Those who best know the needs of the Navy are urging a definite program, to which, unfortunately, some of our legislators are turning deaf ears. We have not, indeed, any silver mines belonging to the Government, but we have the money from taxation, and it has been proposed to distribute a vast sum among certain of our people. Many of us believe that this could be much better employed in carrying out a definite naval program, which would insure

the maintenance of our fleet in a constant state of the highest efficiency. Our statesmen of an older day were very fond of going back to the Greek republics for lessons to guide us. May we not ask those of today to study the story of Themistocles and the upbuilding of the Athenian fleet as an example well worthy of their emulation?

THE ORIGINAL CONTRACT FOR THE *MONITOR*.*

This Agreement made and entered into this Twenty fifth day of October A.D. 1861 by and between Thomas F. Rowland Agent in behalf of the "Continental Iron Works" Green Point Brooklyn of the first part and Captain J. Ericsson of New York Messrs John F. Winslow and John A. Griswold of Troy N.Y. and C. S. Bushnell of New Haven, Connecticut parties of the second part Witnesseth that the party of the first part for and in consideration of a certain sum hereinafter mentioned to be paid to him by the parties of the second part, hereby covenants and agrees to furnish all the tools and facilities, and do all the labor necessary to execute the iron work of an Iron Battery hull, (it being understood that the new ship house now being erected is at the expence of the parties of the second part) said battery to be constructed from the plans and directions which have been or may be furnished the said party of the first part by Captain Ericsson. The party of the first part hereby further agrees to do the said work in a thorough and workmanlike manner and to the entire satisfaction of Captain Ericsson

*Published by courtesy of THOMAS ROWLAND, JR., and CHARLES ROWLAND, sons of the builder of the *Monitor*.

in the shortest possible space of time. And the party of the first part agrees to launch said Battery safely and at his own risk and cost on the East river here and there — delivering her to the parties of the second part. It is also understood that in consideration of the liberal price hereafter stipulated to be paid by the party of the second part that in case the work is not prosecuted with all the vigor and energy practicable then and in that case Captain Ericsson is hereby empowered to instruct the party of the first part to employ a greater number of men or to work a greater number of hours, and which instructions the party of the first part hereby agrees to comply with in order that the work may be completed in the shortest possible space of time as contemplated by this agreement. The parties of the second part hereby agree to furnish all the material for the construction of said Battery delivering the same at the "Continental Iron Work" as soon as practicable after receiving a specification of the materials required for the construction of said Battery. In consideration of the full and faithful — performance of these presents by the party of the first part. the parties of the second part hereby covenant and agree to pay the party

of the first part the sum of Seven and one half (7½) cents per pound (net weight) of iron used in the construction of said hull by the party of the first part Payments to be made weekly in proportion to the progress of the work, the balance remaining to be paid when the hull is launched The parties to this instrument hereby covenant mutually agree that should any alteration in the plans furnished by Captain Ericsson be desired after the same have been executed the party of the first part shall make any alterations that may be deemed desirable by Captain Ericsson at the expence of the parties of the second part. In Witness the hands and seals of the said parties the day and year before written

Witness to the signature of J. Ericsson. C. H. MacBond
Witness to the Signature of J. F. Winslow and John W. Ericsson } F. Ellis

J. Ericsson
J. F. Winslow
John W. Ericsson

Witness to the signature of J. F. Rowland. Warren Estelle

Thos. F. Rowland agt.

A METALLOGRAPHIC STUDY OF FAILURES IN METALS CONDUCTED AT THE U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

By J. F. Daniels, Lieutenant, U. S. Navy.

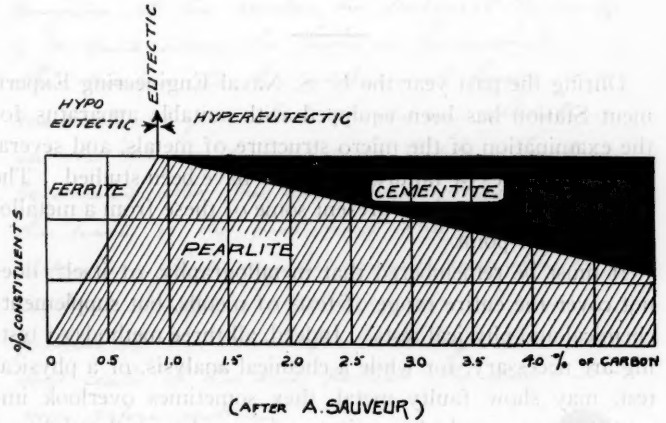
During the past year the U. S. Naval Engineering Experiment Station has been equipped with suitable apparatus for the examination of the micro structure of metals, and several interesting cases of failures of metals have been studied. The object of this paper is to present some of these from a metallographic view-point.

It must be remembered that metallography, of itself, does not cover the entire range of tests of metals, but supplements chemical and physical tests. Indeed, all three methods of testing are necessary, for while a chemical analysis, or a physical test, may show faulty metal, they sometimes overlook important features which are factors in an ultimate breakdown. For the benefit of the many whose fields of labor have not brought them in contact with the study of metallography, an outline of a section of this important science may not be out of place.

Briefly, a metal is made up of a number of irregular polyhedral grains or crystals. These cannot be seen by the unaided eye, but with the metal properly prepared and placed under a microscope the crystalline grains are brought out clearly, and the structure of the metal revealed. The structure thus presented is different for various metals; and unlike in the same metal when the metal has been subjected to a different heat treatment. The structure of ferrous metals depends, in addition to the heat treatment received, on the carbon content. Beginning at the lowest limit, and up to a carbon content of 0.89 per cent., the metal consists of two specific

constituents—ferrite, or pure iron, and pearlite.* At 0.90 per cent. the metal consists of pearlite alone, the eutectic† mixture. Above 0.90 per cent. the metal separates again into two well defined constituents—cementite, or iron carbide, and pearlite.

This relation is diagramatically represented in the following figure (after A. Sauveur):



(AFTER A. SAUVEUR)

A steel containing less than the eutectic proportion 0.90 per cent. carbon, is termed "Hypoeutectic." Above the eutectic proportion of 0.90 per cent. of carbon the steel is termed "Hypereutectic." The eutectic proportion may be considered as the point of carbon content at which the component ferrite and cementite will both be satisfied. Below it the ferrite, or nearly carbonless iron, predominates. Above it the cementite is in excess. Depending upon the carbon content, a metal may be said to be a low, medium, or high-carbon steel. The metals under examination were of the low-carbon variety.

* An intimate mixture, or alternating plates, of cementite and ferrite.

† The eutectic point is the common point of intersection of two inclined branches, and an approximately horizontal line in the freezing-point curves. The alloy which has the composition corresponding with the eutectic point is called an eutectic alloy; before solidification, a eutectic solution; after solidification, a eutectic mixture. (Mellor.)

In making an examination of a metal that has failed in service, certain specimens are selected, prepared and examined. It quite frequently happens that some, or many, of these are discarded and others taken in their stead. When a particular specimen containing a hint as to the probable or contributory cause of the rupture is found, the point from which it is taken becomes a base, and the whole region is carefully examined. This process continues until sufficient evidence is accumulated to establish a logical finding.

If doubt exists as to whether or not the metal has received proper heat treatment, specimens may be compared with standard samples established, or the metal itself may be subjected to proper heat treatment and the structures then compared with records previously made.

EXAMINATION OF A RUPTURED WATER DRUM.

The first failure of metal examined was a steel plate from a section of a water drum of a water-tube boiler that had ruptured in service. Physical tests were made from three tensile-test specimens cut from the ruptured metal, the results of which are given in the following table:—

Original size from which the specimen was taken.	*Ruptured sheet from the top side of crack, from tube sheet.*		
Marked*	1-T	2-T	3-T
Size, in inches	.581×1.5	.5×1.5	.515×1.25
Area, in square inches	.8715	.75	.6437
Length, in inches	8	8	8
Elastic limit, in pounds	32,000	39,000	28,700
Ultimate stress, in pounds	60,890	†52,870	43,490
Elongation, in inches	1.36	1.04	1.5
Fracture, in inches	.4×1.5	.345×1.21	.355×.93
Area of fracture, in square inches	.46	.4174	.33
Elongation, per cent	17	13	18.75
Reduction of area, per cent	47.2	44.3	48.7
Elastic limit, in pounds per square inch	36,700	52,000	44,500
Ultimate stress, pounds per square inch	69,900	70,500	67,500

*1-T cut from end of drum;
2-T cut from center of drum;
3-T cut from back end of drum.

†2-T first broke outside of points at 46,410 pounds.

The samples subjected to tensile tests were marked 1-T, 2-T, and 3-T, respectively. Additional pieces taken from the immediate neighborhood, and given the same identification marks as above, were prepared for microscopic examination.

This work of preparation consists of cutting pieces up into small specimens and polishing the surfaces to be examined to a mirror-like finish. The pieces are then examined under the microscope for flaws or slag, as these are more easily seen at this stage of the work. The next step is to etch the polished specimen with some suitable reagent, such as a nitric or picric-acid solution. This serves to bring out the crystal formation.

The specimen may now be studied under the microscope and, if desired, a photo-micrograph made as a matter of record. Photo-micrograph, Fig. 1, shows a polished specimen. The figure is of low magnification, and shows clearly an extensive slag region. Fig. 2 is a photo-micrograph of the same specimen, but etched to bring out the structure. The magnification here is high. Examination of this figure shows the slag inclusion bordered by great patches of ferrite (the white areas are ferrite, the dark ones pearlite). The grains are abnormally large, and the structure lacks homogeneity. Compare this structure with the finely-balanced formation shown on Fig. 3, a photo-micrograph taken from an excellent piece of metal and selected for comparison. Fig. 4 is a photo-micrograph taken from the same specimen as that used for Figs. 1 and 2, but from a region clear of the slag area. The grains are large and segregation is shown. These specimens were characteristic of the samples marked 1-T.

The samples marked 2-T were next examined. Fig. 5 shows a photo-micrograph taken from one of the specimens prepared. Extreme segregation is evident. This is more clearly shown in Fig. 6, the magnification of which is purposely made low in order to cover a wide field. The streaks or ghosts of ferrite extend throughout the mass, forming planes of weakness therein. As noted in the physical tests, the 2-T metal broke outside the points at 46,410 pounds. This

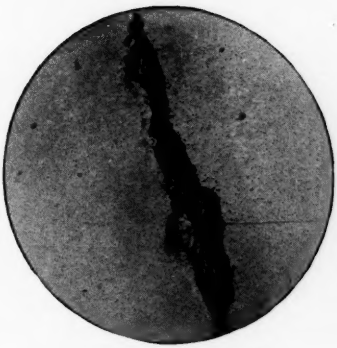


FIG. 1. × 40, not etched.

FIG. 4. × 87, etched with nitric acid.

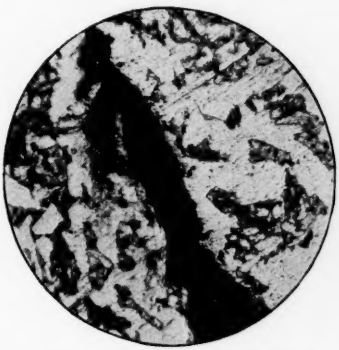


FIG. 2. × 87, etched with nitric acid.

FIG. 5. × 87, etched with nitric acid.

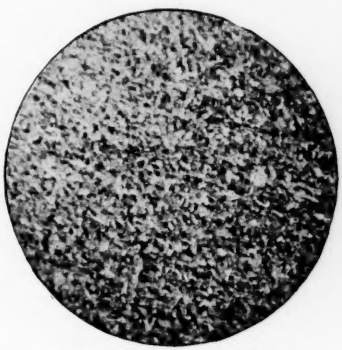


FIG. 3. × 57, etched with nitric acid.

FIG. 6. × 44, etched with nitric acid.

FIG. 7. × 87, etched with nitric acid.

FIG. 10. × 87, not etched.

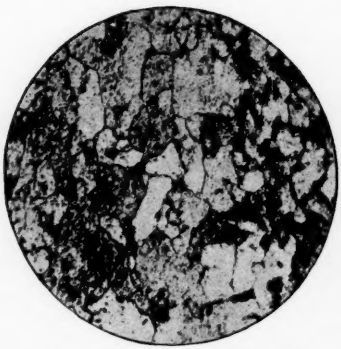


FIG. 8. × 87, etched with nitric acid.

FIG. 11. × 87, etched with nitric acid.

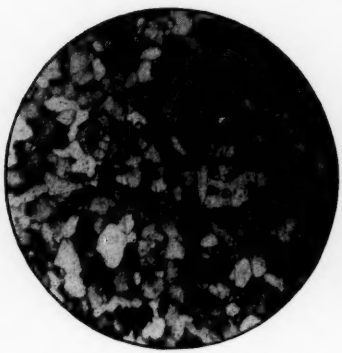


FIG. 9. × 113, etched with nitric acid.

FIG. 12. × 87, etched with nitric acid.

FIG. 13. × 87, etched with nitric acid.

FIG. 16. × 87, etched with nitric acid.

FIG. 14. × 87, etched with nitric acid.

FIG. 17. × 87, etched with nitric acid.

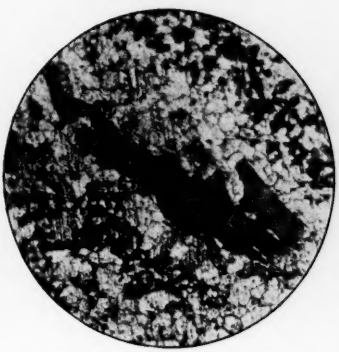


FIG. 15. × 87, etched with nitric acid.

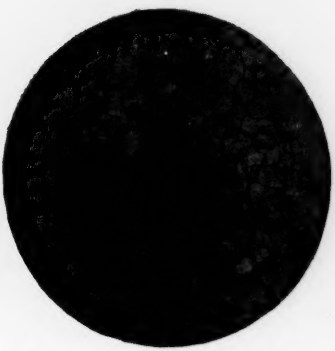


FIG. 18. × 87, etched with nitric acid.

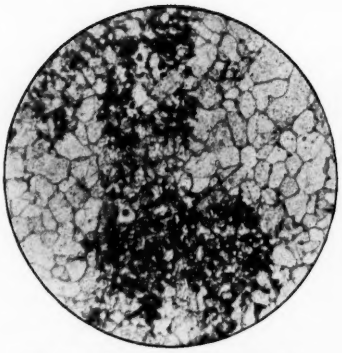


FIG. 19. × 87, etched with nitric acid.

FIG. 20. × 57, etched with nitric acid.

FIG. 21. × 57, etched with nitric acid.

FIG. 22. × 57, etched with nitric acid.

FIG. 23. × 57, etched with nitric acid.

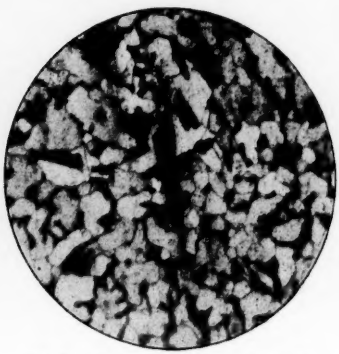


FIG. 24. × 113, etched with nitric acid.

occurred along one of the ferrite streaks or ghosts. These figures are characteristic of the samples marked 2-T.

The 3-T samples were examined, and the results are shown in the following figures: Fig. 7, 8. These photo-micrographs are characteristic of the appearance of the metal mentioned above. Great island-like grains of ferrite predominate. Compare these structures with that shown on Fig. 9, the carbon content of which is very close to that of the metal under investigation. The former lacks, and the latter possesses, homogeneity of structure. There is no doubt that the metal under observation had been subjected to improper heat treatment, as evidenced by the grain growth. The slag areas and the tendency to segregate into streaked regions formed planes of weakness that, ultimately, permitted a breakdown.

BULGED FURNACE OF A SCOTCH BOILER.

The second series of specimens about to be described was taken from the metal of a boiler furnace that had bulged badly in service. The metal is a low-carbon steel.

Figs. 10 and 11 represent the same specimen, the first polished and the second etched. They were selected, from the many pieces examined, to show the great patches of slag inclusion. The metal was filled with impurities, few pieces being found that did not show extensive areas of slag. These areas are shown in photo-micrographic Figures 10 to 17. In nearly every case the slag inclusion is found imbedded in segregated areas of ferrite. Fig. 18 is characteristic of the grain size in regions clear of the large slag areas, while at certain points remote from these regions the structure presented the appearance shown on Fig. 19.

As exhibited in the figures, the metal under observation did not show a degree of fineness of grain indicative of solid and dense structures. The slag areas were extensive enough to form planes of weakness at critical points, being imbedded, or surrounded in every case, by segregated regions of ferrite. Lack of homogeneity is evident throughout, and the internal

structure was weakened by (1) the planes of weakness shown to exist, (2) enlarged grain size, and (3) the abnormal amount of slag present. It would seem that it could only be a matter of time for such a metal to fail in service.

THIRD SERIES.

The third series of specimens examined was taken from the metal of a second ruptured water drum. The metal had passed satisfactory chemical and physical tests, the rupture taking place on final testing after the drum had been assembled. It may be mentioned that certain bending of the metal, to shape the drum, is done cold. Fig. 20 is a photo-micrograph taken from a specimen of this metal. An extensive segregated area was found. The segregation extended along the whole line of the rupture, developing finally into single streaks as shown on Fig. 21. At intervals, minute fissures or cracks were found, as exhibited in Figs. 22 and 23, and distorted grain structures, as exhibited in Fig. 24, showing the effects of cold work on the metal.

PAINTING ENGINE AND FIREROOM BILGES.

By Henry Williams, Naval Constructor, U. S. Navy.

When steel ships and, for that matter, steel buildings came first into general use it was anticipated that especial precautions would have to be taken to protect the steel from corrosion. In the case of ships it was concluded naturally that the outer-skin plating, the part most exposed to the corrosive action of the water, would be the first to deteriorate. As the integrity of the structure depends upon keeping the hull intact, a great amount of care and attention always have been bestowed on that part of the ship. There is another motive in caring for the outer hull of a vessel, and that is the necessity for keeping its surface clear of marine growths and other fouling matter. These two considerations combined have resulted generally in such good care that one practically never hears of a steel ship that has had a hole in its bottom plating due to corrosion from the outside. Frequently holes come as a result of corrosion on the inside. More frequently, perhaps, because nearly all sea-going vessels nowadays have double bottoms, it is the inner bottoms that suffer most from the corrosion, and the locations affected are almost invariably in the machinery spaces. It is a well-known fact that the steel structure in these spaces is especially liable to corrosion and that it is most difficult to keep the surface properly covered with paint. Some ship builders even have adopted the expedient of using wrought-iron plates for inner bottoms in this space, with the idea that it is less subject to corrosion than steel.

Under the electrolytic theory of corrosion, which now is accepted generally and which appears to account for all of the phenomena and to be consistent with observed action, all corrosion of steel surfaces is due to a difference of electric potential between adjacent points. This may be caused by the un-

even distribution of the various components or by the presence of a different metal. Where such difference of potential exists a current will pass when there is an electrolyte present. Moisture is sufficient for this purpose. The substance at the higher potential will pass into solution carrying an electrical charge, which is neutralized by oxidation in the case of steel, and the process continues indefinitely.

The conditions requisite for liability to corrosion are present usually in the locality under discussion. There is almost continual presence of water in the bilges, and it may be hot and salty, or have in solution lye or the soluble salts from ashes, all of which makes it more effective as an electrolyte. Further, it has mixed with it frequently ashes or oil, so that ordinary paint has not much chance of adhering, even if through an unusual effort the bilges are dried and cleaned sufficiently to permit of them being painted at all. The ashes exert an abrasive effect on the paint, and the oil and lye soften and dissolve it. There is often a considerable acceleration of the electrolytic action due to the large amount of brass and copper used in the piping and structures and the frequent presence in contact with the steel of brass and copper filings, or articles of brass stowed in the bilges. The heat of the boilers and of the steam pipes stimulates corrosion, and frequently there is local corrosion due to continued drip of water at some spots.

That serious deterioration has taken place on naval vessels due to such corrosion in the past is evidenced by the bills for repairs to the inner bottoms in machinery spaces. It is the universal experience that at the end of about four years' service it begins to become necessary to patch inner bottoms, and after about ten years the repairs are so extensive as to have cost in the case of a battleship overhauling recently for structural work alone about $15,000.00, besides the incidental costs. These repairs are especially difficult and expensive, due to the necessity for removing piping and auxiliary machinery and to the restricted spaces in which the work must be done.

The writer has had occasion recently to inspect the bottoms in machinery spaces of a number of older vessels in the Navy, and found in every case very marked deterioration of the inner bottom or, on ships having none, of the framing, floor plates and shell plating. This took place in spite of considerable effort of the ship's people to scale and paint the metal, and in several cases there was abundant evidence that the corrosion had proceeded under an apparently sound coat of red-lead paint. In all these cases considerable expenditure of money was necessary for repairs, and in each of them an ounce of proper kind of prevention would have saved many pounds of cure. The Board of Inspection and Survey has had occasion frequently to report unsatisfactory conditions in these spaces and to recommend greater care on the part of the ship's people.

The foregoing should serve to indicate the necessity for heroic measures for the protection from corrosion of steel ships in the spaces mentioned and justify a discussion of the various methods adopted, their advantages and the considerations affecting each.

Formerly the painting instructions for naval vessels required that all metal surfaces below the floor plates in engine and boiler rooms be given at least three coats of red-lead paint made with linseed oil. This method of painting was in vogue for many years, and to say that it has not proved satisfactory states the case mildly.

In the first place, due to the continual presence of moisture of condensation, if not of actual water in the bilges, it is extremely difficult to apply the linseed-oil paint so that it will adhere to the metal and afford the protection it is intended to give. In such cases frequently there is a film of paint apparently sound, but under which corrosion is taking place. In other cases the paint applied has not adhered and the metal was left bare until opportunity came for another trial.

Consideration of the characteristics of paint made from red lead mixed with linseed oil should condemn its use for such locations. Linseed oil is a glyceride of fatty acids and it may

be saponified by the action of water alone. This saponification explains the well-known fact that linseed-oil paints, especially those containing lead, which aids the action, do not resist water. There are, however, many paint pigments and vehicles that do resist the action of water, and special damp-resisting or waterproof paints are made and sold by many firms. Asphaltum products usually are the bases of such paints, China-wood oil and linseed oil treated to eliminate the glycerine being added frequently. There are in existence many paints which, being waterproof, are better suited for the purpose under discussion than red lead and linseed oil. This quality of waterproofness alone does not meet the difficulty of rendering the surfaces dry for a sufficiently long time to permit the application and hardening of the paint. There are damp-proof paints which have the quality of drying in the presence of moisture, and for many locations on shipboard besides those in the machinery spaces such a paint would seem to be almost indispensable.

Dr. Cushman classifies the three main methods generally used for the protection of steel as follows:

(1) Protective coatings of metal and alloys.

(2) Magnetic oxide surfaces.

(3) Paint coverings, including linseed-oil paints, varnishes, lacquers, bitumen and cements.

Under the first class of protective methods, the use of zinc for the protection of steel is the most important and efficacious. Zinc is electro-positive to steel, but its work of protection is accomplished through its own destruction. Bearing this in mind, it is seen readily that even if it were possible to use galvanized plates for inner bottoms, it is probable that the zinc coating would last a short time only. The use of zinc plates, secured to the steel at points especially liable to corrosion, it is believed would prove a valuable expedient. This is done for exteriors of the underwater bodies of naval vessels, and zinc protecting plates are secured around the openings to suctions or discharges the sea chests for which generally are made of brass, and also in the vicinity of the propellers made of bronze.

The second method consists of producing magnetic-oxide surfaces on steel or iron and these are obviously unrustible. Unfortunately the oxide is electro-negative to the steel, and any break in the surface results in more rapid corrosion of the steel due to accelerated electrolysis through the action of the oxide. This method therefore is impracticable for the purpose under discussion.

The third method remains as the one on which main reliance must be placed, and a study of the question of protection of steel inner bottoms reduces itself apparently to a search for the most efficient paint material for the purpose, including, of course, cements under the general heading of paint.

Of all protective coverings, outside of the naval service, Portland cement probably is the one used most commonly for the protection of inner bottoms and bilges. Where it can be made to adhere tightly to the steel it affords excellent protection. It should be borne in mind that there are some kinds of cement which contain calcium sulphate in excess and produce corrosive action on steel, through the breaking down of the sulphate into sulphuric acid.

Unfortunately, however, there are two objections to the use of cement in such quantities as would be necessary here. The first of these precludes it for warships, as its weight is prohibitive when applied in sufficient thickness to form a solid body. The second is the difficulty of obtaining adherence to large flat surfaces, such as inner bottoms of men-of-war, and which are subject to some distortion, either on account of the heat of the boilers or the pressure from the feed tanks underneath when filled with water under head. In spite, however, of its weight Portland cement offers many advantages for use in pockets and spaces that may be difficult or impossible of access for painting. For such purposes considerable quantities are used even on naval vessels.

As has been noted above, among the most efficient of the damp-proof paints are those containing asphaltum as a base, and this class of paints has given excellent service in fire and engine-room bilges. The best known, probably, are the so-

called bitumastic compounds which have been used for a number of years in the merchant service with excellent results. The Bureau of Construction and Repair recognized these facts several years ago and provided in specifications for new ships that bitumastic materials or their equivalent be used on inner bottoms in machinery spaces and on bulkheads and foundations in machinery spaces to the height of the floor plates. The name "Bitumastic" is the proprietary name adopted by one of the firms manufacturing and applying the materials, and seems to have been adopted to describe all similar materials, of which there are a number, apparently differing little in composition or efficacy.

The materials come in three forms, and all of them consist essentially of mixtures of asphalt and coal-tar pitch, thinned when necessary with coal-tar naphtha. The "solution" goes on cold as a first coat and the flats of the bilges are covered then to a thickness of about one-half inch with the cement, which has, in addition to coal-tar pitch and asphalt, some Portland cement. This mixture must be applied hot. The vertical surfaces are covered about one-sixteenth inch thick with the enamel, which consists only of coal tar and asphalt, and also is applied hot.

Formulas for making these materials were developed at the New York Navy Yard, and the results obtained from their use apparently are satisfactory. These formulas are in general as follows:

The "solution" may be made by melting 100 pounds of coal-tar pitch and 50 pounds of Trinidad asphalt. When liquid the mixture is removed from the fire, cooled, and thinners added, consisting of 65 pounds of coal-tar naphtha and 15 pounds of mineral oil.

The "enamel" may be made by melting together 350 pounds of coal-tar pitch and 350 pounds of Trinidad asphalt and boiling for about three hours.

The "cement" may be made by melting together 150 pounds of coal-tar pitch and 120 pounds of Trinidad asphalt and boiling for about three hours, when 150 pounds of Portland cement is added and stirred in.

In all of these care is required in the heating, the quality of the material being dependent mainly on the manner in which it is accomplished. The solution goes on cold as a primary coat in all cases. The surface to which it is to be applied must be scaled and cleaned thoroughly, and be free from dirt, grease or paint. As a final step it is wire-brushed and then brushed with a painter's duster. The solution should not be applied in cold weather when there is any " frost" in the metal. As the fumes are highly explosive the greatest care must be taken in regard to the use of naked lights in the compartment while the material is being applied. It is necessary, also, to provide artificial ventilation in such compartments to permit the men to work to advantage. The priming coat of the " solution" is allowed to stand about forty-eight hours, and while it is still " tacky" the enamel or cement is applied. The cement goes on flat surfaces of bilges, while the enamel goes on vertical surfaces of framing bulkheads, sides of bunkers, trimming tanks, etc. Both, at the time of application, are melted in pots, handled in buckets, and must be worked quickly before they cool. The cement is poured on to the flat surfaces and spread to the desired thickness, about one-half inch. The enamel is applied with a brush having heavy bristles and a handle about fifteen inches long. It goes on in a thickness of about one-sixteenth to one-eighth of an inch. On flat surfaces overhead, such as bunkers, trimming tanks, etc., it is very difficult to apply, and for that reason the solution only ordinarily is used on such surfaces.

The New York yard recently had experience in applying, by the yard force, the bitumastic materials to inner bottoms in machinery spaces on the *Alabama* and *Tacoma*. Considerable difficulty was experienced in securing proper adherence of the materials to the metal, due, probably, to the presence of moisture and oil. Other work progressing in the spaces could not be interrupted, and it was found to be practically impossible to obtain a proper condition of the metal, with correspondingly unsatisfactory results in the adherence of the bitumastic. When orders were received to do the same work

on the *New Hampshire* it was concluded to have the work done by contract, not only in the interest of having it accomplished properly, but to give an object lesson to the yard workmen.

It has been suggested, on account of the softness of the bitumastic cement and its tendency to run on sloping surfaces, that a layer of Portland cement about one-half inch thick on top of the bitumastic will serve a good purpose in protecting from damage and insulating from heat the softer material. Experience with such an arrangement has been had in the case of the *Ajax,* of which the inner bottoms under boilers were coated in 1906, first with an asphaltic cement compound and over that with Portland cement two and one-half inches thick. This was inspected in 1911 and found to be in excellent condition, showing no sign of deterioration of any sort.

In order to test such an arrangement further, for purposes of comparison, the inner bottoms of the *Alabama* and *Tacoma* to which, as was noted above, bitumastic compositions were applied in 1911, had one-half of their surfaces covered with Portland cement about one-fourth inch to one-half inch thick over the bitumastic.

In 1904 the writer inspected the bitumastic materials on the American Line *St. Louis,* which had been applied during construction in 1894–95. Among other parts of the ship the engine and fireroom bilges, foundations, boiler saddles, etc., were protected with the bitumastic. These spaces were in excellent condition and the protective coatings had received only minor attention since first applied. A naval vessel of the same age on which red-lead paint had been used in all probability would have had her inner bottom plating corroded through and have required the expenditure of thousands of dollars in repairs, and during her service would have had much energy and many tons of red lead used up on it.

Experience similar to this was had with the sister ship *St. Paul,* which was protected in the same manner with bitumastic compounds. When this ship went in 1906 to have boilers renewed at the Newport News shipyard, contract was made for

the renewal of the bitumastic in the boiler spaces, as it was anticipated that the work of renewing the boilers would destroy it. Such did not prove to be the case, and the work of repair was confined to replacing the cement that had to be cut out to permit the structural work to proceed.

On the battleship *Arkansas* bitumastic was applied to the trimming tanks before launching, and, in order to trim the ship afterwards, it was necessary to keep the forward ones full of water about eight months. At the end of that time the water was pumped out, the tanks washed down with a hose and the bitumastic was found to be in excellent condition, and required no further work. Had red lead been used the metal undoubtedly would have had to be scaled and repainted throughout after such treatment.

It is readily conceivable that conditions might be such on a ship in commission as to prevent the application of the bitumastic, and in such cases recourse may be had to paint, and one of the damp-proof paints that have been proven in service should be given a trial. There are a number of these, though comparatively few appear to have been tried by actual use on naval vessels due to the predilection for red lead. A damp-proof paint that is probably the best known among architects and builders of structural steel is the " Tockolith cement paint," which not only is damp-proof, but has the desirable quality of drying even when applied to moist surfaces. This paint is made up with a pigment which resembles closely Portland cement, and is dependent on China-wood oil for its damp-proof qualities.

Another paint that perhaps is better known among naval officers, due to the excellent results it has given when applied to trimming tanks of submarines, is the " Rabok liquid-carbon" paint. This consists of a graphite pigment with metallic lead, and an asphaltic oil thinned with benzine.

Another paint which offers promise and for which its proprietors claim damp-proof qualities, added to the ability to adhere to moist steel and afford protection, is " Minwax." This is made from a mineral bituminous wax mined in Oklahoma,

and thinned with bituminous oil distilled from the same natural substance.

"Lubrose" paints, which are made in Germany, have been tested and afforded good protection to the bilges of yard craft on which tested.

"Eureka" paint which has given good service, where applied for test as a substitute for red lead, is made with a specially treated linseed oil, from which in all probability, the gylcerine has been removed to render it damp proof. The pigment is similar to venetian red, containing iron oxide, calcium sulphate and silica.

The "Bitumo" paint has the same general characteristic as the bitumastic materials except that it can be applied cold with a brush like ordinary paint. On test it afforded excellent protection to steel plating in spaces where water was present continually.

While there are many other paints possibly as good or even better for the purpose than the ones mentioned, the writer's practical experience has been limited to these.

For ships already in service and on which red lead has been used in bilges, bunkers, trimming tanks and elsewhere, the best means of solving the difficulty apparently is to have recourse to a damp-proof paint until favorable opportunity offers for applying the bitumastic materials.

Another feature in connection with the use of red-lead paint that recently has been attracting attention is the question of lead poisoning, which may be caused by its use. There have been numerous undoubted cases on shipboard and at navy yards, and precautionary measures are necessary and desirable to eliminate the possibility of serious effects on the men using lead paints or scaling surfaces that have been painted with them. The last phase of the question was regarded as being of enough importance to warrant the issuing by the Bureau of Medicine and Surgery of a special circular letter to all medical officers on the subject of lead poisoning, detailing its causes, symptoms and the precautions to be observed to prevent it.

REPORT OF TEST OF A MURPHY STOKER ON A BABCOCK & WILCOX BOILER, AT THE NAVAL GUN FACTORY, WASHINGTON, D. C.

On February 14th and 15th, 1912, a Board composed of Captain H. P. Norton, Lieutenant Commander D. E. Theleen, Lieutenant H. Frankenberger, and Lieutenant L. H. Irwin, U. S. N., conducted a twenty-four-hour efficiency test, a four-hour test at 50 per cent. overload, and a two-hour capacity test, on a Murphy Stoker in boiler No. 16 at the power house of the Naval Gun Factory.

The twenty-four-hour test ended at 10:00 A. M. February 15, and the four-hour 50 per cent. overload test was started immediately. This ended at 2:00 P. M. February 15, and the two-hour capacity test started immediately thereafter, the test being finished at 4:00 P. M. February 15th.

On the completion of each of the tests the fires were practically in the same condition as at the beginning of that test. The conditions at the beginning and ending of each test with regard to height of water and the condition of coal in magazines were the same.

The *operation* of the furnaces required the services of at least two men, one on top of the hopper to distribute the coal into the magazines (with coal chutes arranged to deliver the coal into the magazines this man would not be required), and one in front of the furnace to operate rakes, slice bars, and to agitate the stoker by hand. The slice bars were operated about every 10 minutes during the twenty-four-hour test. This was called "rousting." The stokers were agitated with a large wrench about once an hour during the twenty-four-hour test.

During the four-hour overload test, these operations were made more frequently. The operator used the rake for distributing the coal on the grate.

During all tests, boilers 11 to 15 on the west stack (Wilkerson stokers burning buckwheat coal) were lighted but fires were banked. Boilers 16, 18 and 20 with Murphy Stokers were in operation. Boiler No. 17, adjacent to No. 16, was dead.

SMOKE.

All smoke observations therefore gave smoke emitted from the four boilers with Murphy Stokers. It was found that when the fires were raked or sliced the amount of smoke was increased.

During the twenty-four-hour test, 73 observations were taken between 1:48 P. M. and 6:11 P. M. on February 14th, 1912, this giving an average of one observation every 3.6 minutes. The average duration of smoke was 35.2 seconds. On February 15th, from 7:17 A. M. to 10:00 A. M., 41 observations were made with an average with one observation every 3.6 minutes. The average duration of smoke was 39.6 seconds.

The longest periods were as follows:

2:25 P. M., Feb. 14, 90 seconds. Started heavy and dark; soon getting lighter.

4:28 P. M., Feb. 14, steady 90 seconds, interval of a few seconds, then steady 55 seconds.

4:40 P. M., Feb. 14, 15 seconds light, 45 seconds dark, 60 seconds light.

5:23 P. M., Feb. 14, 120 seconds dark followed by 120 seconds light. This was caused by raking the fires necessitated by man on top of hopper allowing one corner of magazine to get empty.

6:10 P. M., Feb. 14, 90 seconds dark, 70 seconds light.

7:22 A. M., Feb. 15, 2 minutes black smoke.

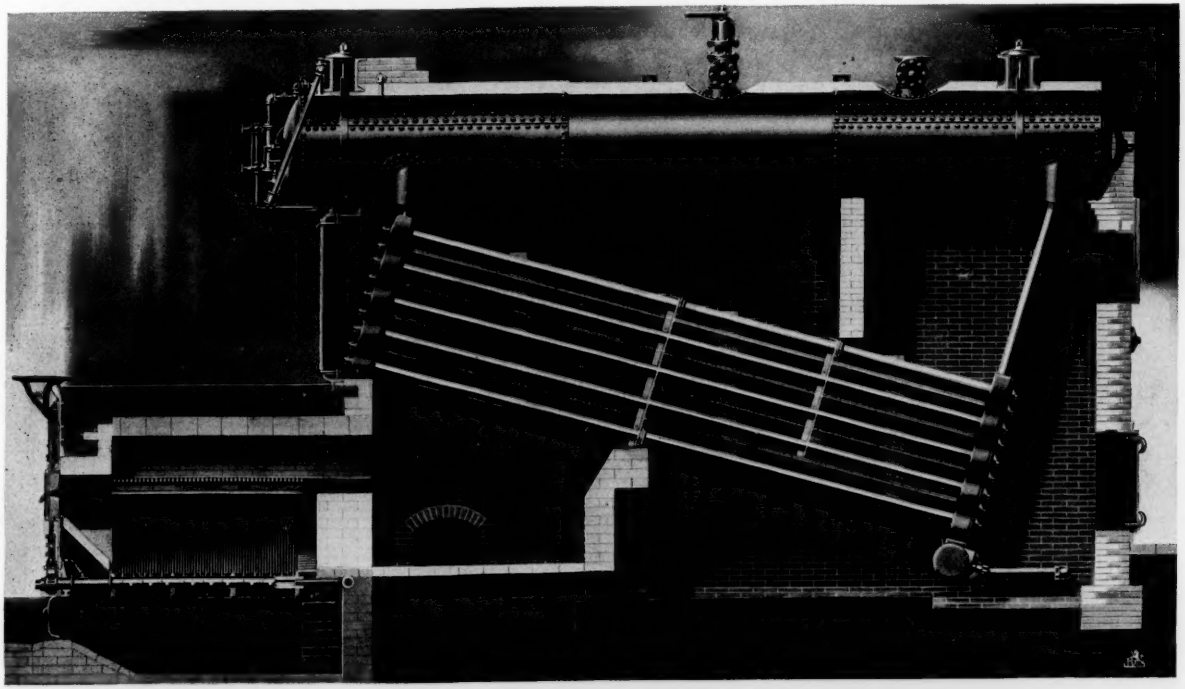
7:33 A. M., Feb. 15, 1 minute black smoke.

7:52 A. M., Feb. 15, 1 minute light smoke.

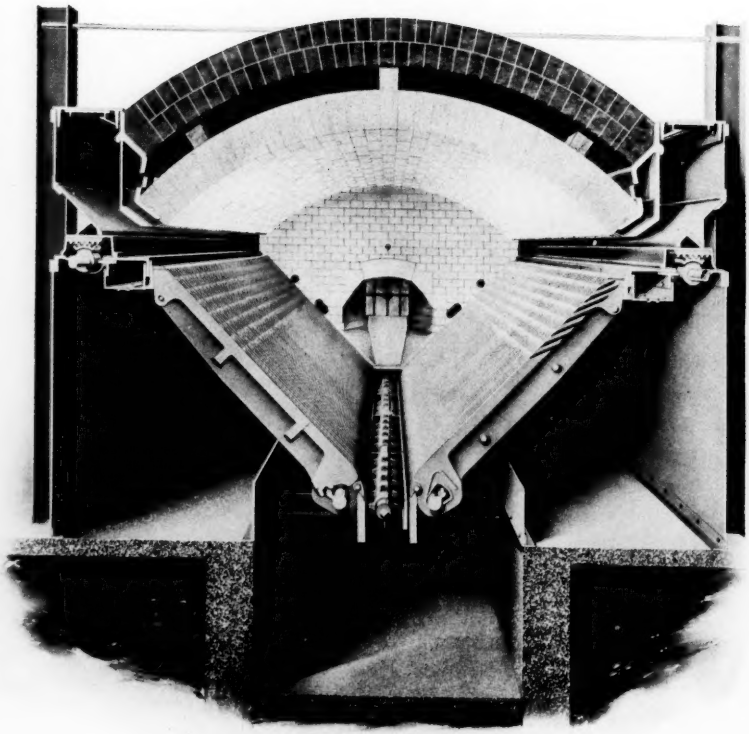
9:07 A. M., Feb. 15, 1 minute 45 seconds strong steady flow. (See Photographs.)

9:14 A. M., Feb. 15, 1 minute 20 seconds, commencing dark and getting lighter.

9:30 A. M., Feb. 15, 120 seconds (Strong 60 seconds, light 60 seconds).



LONGITUDINAL SECTION THROUGH BOILER AND FURNACE.



TRANSVERSE SECTION THROUGH GRATE.

Four-hour overload test:

Fifty observations were taken, averaging one observation every 4.8 minutes. The average duration of smoke was 32 seconds. Longest periods were:

10:17 A. M., Feb. 15, 90 seconds.

10:30 A. M., Feb. 15, 20 seconds.

10:45 A. M., Feb. 15, 18 seconds black smoke followed by 12 seconds light smoke.

10:57 A. M., Feb. 15, 45 seconds black smoke followed by 60 seconds of light smoke.

11:46 A. M., 90 seconds alternating dark and light.

The two-hour capacity test:

Sixteen observations were taken from 2:00 to 3:00 P. M. The average duration of smoke was 75.9 seconds. Longest periods were:

2:22 P. M., 40 seconds medium flow and color; 10 seconds no smoke; 1 minute 50 seconds, medium flow and clear; 3 seconds no smoke; 50 seconds, light smoke; 5 seconds, no smoke; 40 seconds, light smoke; 5 seconds, no smoke; 30 seconds, light smoke.

2:29 P. M., 2 minutes dark smoke; 1 minute light smoke followed by 30 seconds dark smoke; 5 seconds, light smoke; 30 seconds dark smoke; 15 seconds light smoke; 15 seconds dark smoke; 30 seconds light smoke.

2:40 P. M., 24 seconds dark smoke; 15 seconds light smoke; 10 seconds dark smoke; 15 seconds light smoke; 10 seconds dark smoke; 20 seconds light smoke; 10 seconds dark smoke; 70 seconds light smoke.

2:48 P. M., 2 minutes 30 seconds steady flow light smoke.

2:55 P. M., 50 seconds medium color; 45 seconds light smoke; 20 seconds black smoke.

During the four-hour test and the two-hour test, it is noted that smoke was emitted in puffs, this being due, it is believed, to the slicing and raking of the fires.

The smoke laws of the District of Columbia, approved February 2d, 1899, state in part as follows:

"The emission of dense or thick black and gray smoke or cinders from any smoke stack or chimney used in connection with any stationary engine, steam boiler, or furnace of any description within the District of Columbia shall be deemed and is hereby declared to be a public nuisance."

The Murphy Iron Works guarantees that the emission of smoke will comply with the smoke laws of the District of Co-

lumbia and they will assume liabilities for one year for any penalties arising therefrom. The Board reported that the emission of smoke during the tests complied sufficiently close to the smoke laws of the District to warrant the accepting of stokers with regard to this item.

It was impracticable to conduct a test to determine the consumption of the Acme engine which drives the Murphy Stoker. This is a two-cylinder single-acting automatic-cut-off engine. Diameter of cylinders, 4 inches; stroke, 5 inches. The revolutions during the test averaged from about 340 to 485 per minute. From data received from other tests on the engine it is believed that the engine did not consume as much as ¾ of 1 per cent. of the total steam generated.

SIFTINGS.

The siftings which fell into sifting pits were raked out every four hours and weighed, with the following results:

From 10:00 A. M. to 2:00 P. M., February 14, 1912, estimated about 130 pounds.
From 2:00 P. M. to 6:00 P. M., February 14, 128 pounds.
From 6:00 P. M. to 10:00 P. M., February 14, 106 pounds.
From 10:00 P. M., Feb. 14, to 2:30 A. M., February 15, 100 pounds.
From 2:30 A. M. to 6:00 A. M., February 15, 84 pounds.
From 6:00 A. M. to 10:00 A. M., February 15, 113 pounds.
From 10:00 A. M. to 2:00 P. M., February 15, 90 pounds.
From 2:00 P. M. to 4:00 P. M., February 15, 68 pounds.

These siftings were thrown back into the hopper, entailing no loss of coal.

CLINKER.

At 10:00 P. M. February 14th, 1912, 20 minutes was required to remove clinker from grates, this work being done by the Murphy Company's firemen.

At 6:30 A. M. February 15, 11 minutes was required by two of the Murphy's men to remove the clinker. Total amount of clinker removed during the twenty-four-hour test, 383 pounds.

At 1:50 P. M. February 15, 10 minutes was required to remove clinker. Total amount of clinker removed, 105 pounds.

After the completion of the four-hour test an examination was made of the boiler and furnace, and the furnace and stoker were found to be in excellent condition.

The results of the evaporation tests were as follows:

DESCRIPTION AND DIMENSIONS OF BOILER AND APPURTENANCES.

Type of boiler, Babcock & Wilcox water tube.
Drum, diameter, 36 inches; length, 23 feet 3½ inches.
Tubes, number, 140; diameter outside, 4 inches; length, 18 feet; thickness, 10 gage.
Furnace, kind of, Murphy stoker.
Grate surface, length, 7 feet; width, 7 feet; area, 61 actual square feet, projected square feet.
Heating surface, area, 3,000 square feet; ratio to grate, 49.2 to 1 actual, 61.2 to 1 projected.
Per cent. water-heating surface, 90.4; per cent. superheating surface, 9.6.
Smoke pipe, area at damper, 12 square feet.
Water space, 357 cubic feet; steam space, 141.3 cubic feet.
Stoker engines, kind, semi-single-acting, twin-cylinder; dimensions of cylinders, 4 inches × 5 inches.
Feed heater, kind, 2 Warren Webster open feed-water heaters.
Economizer, kind, Green-fuel economizer; area of surface, 1,728 square feet.
Feed pumps, kind, Worthington Duplex; dimensions of cylinders, 16 inches × 10½ inches × 10 inches.

DESCRIPTION OF APPARATUS.

Method of weighing water, one tank on platform scales. (Scales were tested by sealer of weights and measures before test.)
Method of weighing fuel, platform scale, which was tested by sealer of weights and measures.
Kind of calorimeter used, none. Superheated steam.
Method of taking temperature of and sampling flue gases, thermometers and pyrometers; Uehling, Hayes, and Orsatt CO_2 instruments.
Condition of boiler before and after test, boilers and furnaces clean before test. On completion of test, the stoker and furnace were found in excellent condition.

Number of test	1	2	3
Date of test, 1912	Feb. 14–15	Feb. 15	Feb. 15
Duration of test, hours	24	4	2
Kind of fuel	New River (white oak).		
start	Flying.	Flying.	Flying.
State of weather	Fair.	Overcast.	Overcast.

	1	2	3
Average pressures:			
Thermometer attached, degrees F....	72.9	75.0	74.8
Barometer corrected for temperature	30.26	30.06	29.98
in pounds.....	14.84	14.75	14.71
Steam pressure by gage, pounds.....	124.84	128.87	131.5
absolute, pounds.....	139.68	143.62	146.29
Force of draft at base of pipe, inches of water.....	.92	.91	.90
Force of draft inside of damper	.39	.67	.73
in furnace.....	.14	.26	.28
Revolutions of stoker engine	342.73	372.78	485.00
Complete cycles of engine per minute..	.43	.47	.61
Average temperatures:			
External air, degrees F.....	33.63	32.56	34.8
Fireroom, degrees F.....	84.69	88.89	89.2
Steam, degrees F.....	434.71	457.67	459.00
Feed water entering heater, degs. F..	35.98	36.00	36 00
economizer, °F..	108.24	109.78	101.60
boiler, degs. F..	184.45	195.67	194.20
Air entering ash pit, estimated.....	80.00	83.00	83.00
Escaping gases from boiler, degs. F..	497.76	607.11	622.00
to economizer, °F....	423.19	522.78	527.00
from economizer, °F.	270.93	336.67	339.00
Fuel:			
Kind of.....	New River (white oak).		
Weight of coal as fired, pounds.....	24,471.00	6,143.00	4,038.00
Moisture in coal, per cent	1.77	3.28	2.88
Weight of dry coal consumed, lbs...	24,038.00	5,942.00	3,922.00
Weight of ash and refuse, pounds...	1,460.00	452.00	406.00
Weight of combustible consumed, pounds	22,578.00	5,490.00	3,516.00
Per cent. of refuse in dry coal.....	6.07	7.61	10.35
Fuel per hour:			
Coal consumed per hour, pounds.....	1,019.6	1,535.8	2,019.00
Dry coal consumed per hour, lbs.....	1,001.6	1,485.5	1,961.00
Combustible consumed per hour, lbs.	940.7	1,372.5	1,758.00
Coal consumed per hour per square foot projected G. S., pounds	20.81	31.34	41.20
Dry coal consumed per hour per square foot projected G. S., lbs.....	20.44	30.32	40.02
Combustible consumed per hour per square foot projected G. S., lbs.....	19.20	28.01	35.38
Coal per hour per sq. ft. H.S., lbs...	.340	.512	.673
Dry coal per hour per sq. ft. H.S., lbs.	.334	.495	.654
Combustible per hour per square foot H.S., pounds	.315	.457	.586

	1	2	3
Quality of steam:			
Per cent. of moisture in steam	0	0	0
Degrees of superheating	81.80	102.60	102.53
Water:			
Total weight of water fed to boiler, corrected	258,742.00	55,766.00	33,884.00
Factor of evaporation	1.1192	1.1195	1.1214
Equivalent water evaporated into steam from and at 212 degrees, pounds	289,589.00	64,230.00	37,998.00
Water per hour:			
Equivalent evaporation from and at 212 degrees	12,066,00	15,608.00	18,999.00
Equivalent evaporation from and at 212 degrees per sq. ft. G. S., lbs.	246.2	318.5	387.7
Same per sq. ft. of heating surface, lbs.	4.02	5.20	6.33
Economic results:			
Water apparently evaporated under actual conditions per pound of coal as fired, pounds	10.57	9.08	8.39
Apparent equivalent evaporation from and at 212 degrees per lb. of coal (including moisture), pounds,	11.83	10.16	9.41
Equivalent evaporation from and at 212 degrees per pound of dry coal, pounds	12.05	10.51	9.69
Equivalent evaporation from and at 212 degrees per lb. of combustible, pounds	12.83	11.37	10.81
Efficiency:			
Efficiency of boiler; heat absorbed by the boiler per lb. combustible divided by the heat value of one lb. of combustible	78.61	69.33	65.77
Efficiency of boiler, including grate; heat absorbed by the boiler per lb. of dry coal, divided by the heat value of one lb. of dry coal	77.46	67.36	61.91
Remarks and observations:			
Principal data taken every half hour:			
Percentage of smoke as observed		Photographs.	
Method of firing		Mechanical stoker.	Natural draft.
Boiler horsepower by steam-flow meter	355.51	*485.00	*614.00
Boiler horsepower by computation	349.7	452.4	550.7
Per cent. of rating (300 B.H.P.)	116.6	150.8	183.6
Coal per hour per B.H.P.	2.91	3.39	3.67

*Too few observations to be correct.

Fuel and Gas Analyses.

Proximate analysis of fuel: Test	No. 1 (24 hrs.) Coal.	Comb.	No. 2 (4 hrs.) Coal.	Comb.	No. 3 (2 hrs.) Coal.	Comb.
					Per cent	*Per cent.*
Fixed carbon	73.73	78.32	73.16	78.50	73.10	78.32
Volatile matter	20.40	21.68	20.03	21.50	20.24	21.68
Moisture	.90	...	1.71	...	1.61	...
Ash	4.97	...	5.10	...	5.05	...
Total	100.00	100.00	100.00	100.00	100.00	100.00
Sulphur separately determined	.85	.91	.85	.91	.91	.97
Ultimate analysis of dry fuel:						
Carbon (C)	85.10	89.29	85.20	89.56	85.35	89.64
Hydrogen (H)	4.74	4.97	4.76	5.00	4.76	4.99
Oxygen (O)	3.29	3.45	3.00	3.15	2.83	2.97
Nitrogen (N)	1.32	1.39	1.32	1.39	1.36	1.43
Sulphur (S)	.86	.90	.86	.90	.92	.97
Ash	4.69	...	4.86	...	4.78	...
Total	100.00	100.00	100.00	100.00	100.00	100.00
Moisture in sample of fuel as received	1.77	...	3.28	...	2.88	...

Analysis of ash and refuse: Test	No. 1. Clinker.	No. 2. Clinker.	No. 1. *Per cent.*	No. 2. *Per cent.*	No. 3. *Per cent.*
Carbon	6.30	5.23	55.15	58.55	71.02
Earthy matter	93.70	94.77	44.85	41.45	28.98

Calorific value of fuel: Kind of calorimeter used: Peters Mahler. Test	No. 1. *B.T.U.*	No. 2. *B.T.U.*	No. 3. *B.T.U.*
Calorific value by calorimeter, per lb. of dry coal	15,001	15,039	15,013
Calorific value by calorimeter, pr. lb. of combustible	15,792	15,835	15,825
Calorific value by analysis, per pound of dry coal	15,095	15,140	15,188
Calorific value by analysis, per lb. of combustible	15,838	15,914	15,950
Analyses of dry gases: Test	**No. 1.** *Per cent.*	**No. 2.** *Per cent.*	**No. 3.** *Per cent.*
Carbon dioxide (CO_2), last pass (Orsatt)	12.57	12.34	13.68
Oxygen (O), last pass	*6.69	6.76	4.50
Carbon monoxide (CO), last pass	* .03	...	.58
CO_2 back connections (Uehling)	11.13	8.99	10.10

* 22 observations.

GEORGE WALLACE MELVILLE.

The death of Admiral Melville brought sudden ending to a career which was marked by noble achievement in many fields. Beginning as a young and daring officer of the Navy during the Civil War, he became in later years the "lion-hearted Melville" of Polar Seas, the leader and hero of a march and search which are without parallels in all Arctic history. Still later, as Engineer-in-Chief, he gave to the Navy sixteen brilliant years of most strenuous labor during the critical period of passage from the old to the new régime in naval engineering. Then, although retired from active duty, his life work still went on unceasingly. There was no lessening of his devotion to his profession, to the Navy, and to the flag he loved, until, full of years and honors, he passed so swiftly away on March 17, 1912.

The home of the family from which Admiral Melville was descended, is at Stirling in Scotland, the gateway to the Highlands, near the field of Bannockburn, and the spot in which Sir John Melville in 1549 died on the scaffold for his faith. The long line of the Melvilles has held an honored place, from early days, in Scottish song and Scotland's history; and, in this later time, the stout old race loses no glory through such men as he. In view of his naval and Arctic record, it is interesting to note that, but two generations from him, we find, in his family, Sir James Melville, First Lord of the British Admiralty when the expedition of Parry to Baffin's Bay was despatched, and whose name has been given through this to some Arctic lands and waters.

In 1804, James Melville of Stirling landed in New York City, bringing with him a large family of sons, among whom were James and Andrew—historic Melville names—George and Alexander. The latter was the father of Admiral Mel-

ville; and, at sixteen years of age, he was sent again across the sea to complete his education at the old University of St. Andrews, which had trained so many of his line. Returning to New York, he became a professional chemist, with reference especially to dyeing and the allied arts, and acquiring a competence thereby. He was widely known and liked and was a power in the politics of his day. During the American Civil War, he filled to the full the measure of patriotism, giving three sons to the service of the Government and raising and equipping a company of volunteers.

He married Sarah Douther Wallace of New York, and became the sire of stalwart sons. In stature, he and they were of the Anakim, and any one of them could have swung with ease, in cut and thrust, the great two-handed sword of William Wallace which hangs today in the armory of Dumbarton Castle. The father—"Big Sandy" as he was called—was six feet, six inches tall; one son was six feet four; another six feet two; and George, the oldest, was but a shade under six feet.

The latter, the distinguished admiral of later years, was born on January 10, 1841, in the city of New York. He was educated in its public schools and later, displaying marked mechanical instincts, passed through the Polytechnic school of Brooklyn and received a further course in mathematics in a religious academy of that city, which, in that branch, held high repute. With the hard-headed wisdom of his race, the father resolved that his son should be well grounded in the profession to which his tastes and later training tended; and so, when school days were ended, the lad was entered in the engineering works of James Binns of East Brooklyn, where Melville laid broadly and deeply the foundations of that practical skill which served him so well in many a later hour of trial.

On April 12, 1861, Southern guns were trained on Sumter —the first argument of force to prove that the United States was not a nation. The great wave of loyalty which surged over the startled North could not but sweep Melville with it. He was in the dawn of manhood; devoted to the land of his

birth; and there stirred within him as well the blood of the stern race from which he sprung—that race whose wild "pipes of the misty moorlands" have sounded above so many trampled battle sods the martial pibroch, "heard so often as earth's latest music by dying men."

His training and an inborn love of adventure, turned him toward engineering duty and to the sea; and, on July 29, 1861, when but twenty years and six months old, and but ninety days after the war began, he became an officer of the engineer corps of the United States Navy. It was well, indeed, for the nation that, in its extremity, it could command the services, in sufficient number, of men of his profession; for, in the struggle which was at hand, the naval engineer was to play a vital part.

On March 4, 1861, the North was in possession of but two forts—Sumter and Pickens—on the Atlantic and gulf coasts below Chesapeake Bay. There were three thousand miles of sea coast to be blockaded, necessarily by steamers; and, finally, if this blockade were not early made effective, foreign governments stood ready, after but brief delay, to demand the independence of the Confederacy, with whose people their rulers were in sympathy, and for whose great yield of cotton the mills of the world were waiting.

To meet this emergency, the Navy Department had available but sixty-nine vessels of all classes, of which thirty-four were sailing ships. Fortunately, indeed, the tireless genius of Gideon Welles, the greatest of naval secretaries, swayed the helm of the Department, and by him the increase of the fleet was pressed so energetically that, at the close of the year 1861, through the purchase and construction of steamers of every class, the Government floated a navy of 211 vessels, bearing 2,301 guns, and carrying 20,000 men, although of these the ships of greatest tonnage and battery power were neither steamers nor modern vessels. At the close of the war the force consisted of about 600 ships of all classes, including 100 in service on the Western rivers.

In looking backward through those years, it seems difficult

to overestimate the value of the great steam fleet to the Northern arms. Through its early and complete blockade and through the unbroken tenure of Fort Pickens, the Government was enabled, in the beginning, to dispute successfully the claim of the Confederacy to possession of the Southern sea coast and to consequent recognition as a nation. Later on, and to the end, there were taken or destroyed by that fleet 1,504 vessels of all classes, having a value, with that of their cargoes, of at least thirty-one millions of dollars.

The blockade was then one of the main factors in ending the Civil War. While the deeds of the Navy on inland waters and the victories of the *Kearsarge* and *Wachusett* on foreign seas, make a record whose superior in skill, resource, and daring, the ages cannot show, yet it is to that great fleet, lying in long line off the low Southern shore, that history will turn to seek the Navy's cardinal service in that time of trial.

Of this, Professor Soley says: "As a military measure, the blockade was of vital importance in the operations of the war." Through it, supplies from abroad were cut off, King Cotton reigned in vain, and the South and its armies, despite their gallant struggle, were starved into surrender. To achieve these results, within four years, if at all, without it and with that sunlit sea coast open to the world, would seem to have been impossible.

That great blockading fleet would have seemed, in 1861, an idle dream to Gideon Welles, if he had not known that it was to consist, not of massive battleships, but of small and handy steamers whose chief qualities should be speed and sea-keeping powers. Its strength lay, then, in these and not in armor nor in ordnance; and the backbone of that strength was the skill and patriotism of the engineers of the North. If they had not been ready at the nation's call and had not come, as Melville came, the war would have been either greatly prolonged or, with foes so valiant and determined, aided from without, it might have ended swiftly in disaster.

Melville, although but a junior officer during the war, saw much active service, and faced afloat the stress and shock of

conflict in not a few engagements. He was for a brief period on the side-wheel steamer *Michigan,* cruising on the Northern lakes, and was transferred thence to the screw sloop-of-war *Dakotah* of the North Atlantic fleet. With her he served at the shelling of Lambert's Point and the capture of Norfolk, Va., she acting as guard ship during the night and morning of the destruction of the *Merrimac* off Craney Island, Norfolk Harbor.

He was then with the fleet that cleared the James River, in Virginia, as far as Fort Darling, Drury's Bluffs, and participated in the long engagement which attended the effort to reduce that work. He served still further in the James with the fleet which covered the retreat of McClellan's army to Harrison's Landing.

After Farragut had made his brilliant passage beyond Forts Jackson and St. Philip and up the Mississippi river, the *Dakotah,* to which Melville was then attached, carried dispatches, to the admiral to proceed still further up the river. In returning from this service, Melville was stricken with typhoid fever, contracted in the James, and was transferred to the hospital at Key West, Fla.

After convalescence, he was attached to the fast side-wheel steamer *Santiago de Cuba,* and, leaving her, joined at sea the screw sloop *Wachusett,* nine guns, 1,032 tons. This vessel proceeded to Philadelphia for repairs, and, pending their completion, Melville served temporarily on the steamer *Tonawanda,* chartered to pursue the Confederate cruiser *Tallahassee.* Rejoining the *Wachusett,* Captain Napoleon Collins, he went in that ship to Brazil, where she was ordered to intercept Confederate cruisers. She remained at sea on this duty, near the equator and off the Brazilian coast, for a considerable period, during which, it is pleasant to note, that the ship, though but a small sloop-of-war, never sighted a sail which she did not overhaul,—a record which speaks volumes for the efficiency of her engineering staff.

While refitting in the harbor of Bahia, Brazil, the *Wachusett* became the actor in one of the most striking and dramatic

engagements of the war,—the capture of the Confederate cruiser *Florida* in the neutral waters of that port. The latter vessel was built at Liverpool in 1862 on the model of British gun vessels. She carried a battery of two 7-inch rifles and six 6-inch guns, and, in her brief career, had done much damage to American merchant shipping.

She came into the outer harbor of Bahia on the night of October 5, 1864, revealed her identity boldly when hailed, and, shortly thereafter, although the decks of the *Wachusett* were cleared for action, moved to an inner berth within five-eighths of a mile of that vessel. Between the two, to keep the peace, lay a Brazilian corvette, and there were in addition, ready for this service, a 22-gun sailing sloop and the forts on shore, one of 46, the other of 16 guns.

The *Florida* had received permission to remain for forty-eight hours, during which Captain Collins, of the *Wachusett*, challenged formally the captain of the cruiser to a naval duel, with their respective vessels, beyond the marine league. This the latter officer declined and Collins then determined, before the time limit expired, to capture or destroy the cruiser in the harbor. As such an act was contrary to international law and might be attended with much hazard in the guarded port, Collins called, in his cabin, a council of war, at which Melville, with other officers, was present, to consider the surest mode of attack.

Many methods were discussed, but the choice fell upon that suggested by Melville—to ram the *Florida* where she lay. It was contended by more than one member of the council, that, when the *Wachusett* struck her blow, the shock would start the boilers from their fastenings, thus breaking steam joints and scalding fatally all below. Melville combated this theory and ended the discussion upon it when he said to the captain:

"I do not think the boilers will break loose; but, if they do, there need be but one man sacrificed, for, after the engines are started, I can work them alone and will order all hands on deck."

While preparations for the attack were making, Melville

volunteered to board, in *mufti*, the *Florida* in broad daylight, and ascertain, if possible, the strength of her battery and the location of her machinery. If, in this mission his identity were discovered, he might have been strung up, as a spy, to the yard-arm; but, notwithstanding this danger, the fearless young officer succeeded in reaching the enemy's gangway in a shore boat, gained her deck, and with a quick glance fore and aft obtained the desired information before the officer in charge hustled him over the side and into his boat with the injunction that, if he did not keep off, a shot would be dropped through her bottom, leaving him to sink or swim in the bay.

At 2 A. M., on October 7, the *Wachusett* began her brief but hazardous voyage. Melville, however, was not entirely alone below, one fireman, Bradley, having refused to leave him. She struck the *Florida* on the starboard side, abaft the beam, and cut her down for eighteen inches below the water line, carrying away her main yard and mizzen mast, throwing her after broadside gun down the cabin hatch, and burying her ship's company below the awnings, which were forced down on them by the wreckage.

A boarding party from the *Wachusett* surged on her from forward; there were a few pistol shots from the *Florida* which were answered by a volley from the Union vessel and the discharge of broadside guns; and the victory was complete. A hawser was made fast to her foremast and the *Wachusett* towed her off to sea, running, on the way, the gauntlet of a *pro forma* fire from the Brazilian forts and vessels-of-war. The action lasted but twenty minutes, and there were but three men wounded, a bugler, the engineer's yeoman, and Melville himself, who, after the cutting out, received a severe axe stroke across the back of the left hand.

The *Wachusett* proceeded with her prize to Hampton Roads, Va.; and there the *Florida* was sunk, according to the official declaration of the United States Government, through "an unforeseen accident," after a collision with an army transport —which "accident," it may be remarked, was most opportune

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in its final settlement of the question as to the custody by the United States of the Confederate cruiser.

Leaving the *Wachusett,* Melville volunteered, in answer to Porter's stirring and famous call, in which the admiral used the words, "Death, glory and promotion." He was assigned to torpedo boat No. 6 and served with Porter's fleet at the capture of Fort Fisher, and subsequently in the hazardous duty of clearing and buoying the channel of the Cape Fear river, that the fleet might advance to Wilmington, N. C. Later he joined the gunboat *Maumee* in the James river, which little craft was the first vessel to cut and ram her way up the James as far as the Rockets below Richmond, the capital of the Confederacy, on the day of its evacuation and while the city was still burning.

When the war had ended, Melville was ordered to the *Tacony,* and, with her, served in the Mexican gulf during the French occupation and evacuation of Mexico; later he joined the gunboat *Penobscot;* then cruised to Brazil in the flagship *Lancaster;* then to the Arctic in the steamer *Tigress;* then to China and Japan in the flagship *Tennessee;* again to the Arctic in the steamer *Jeannette;* and yet again in the *Thetis* for the relief of the Greeley expedition. The latter duty, with a short term on the *Atlanta,* closed his service afloat, which, after the war, was interspersed with brief details to the various navy yards.

It is scarcely necessary to note that, during his cruising, Melville was commended officially in more than the usual form by commanding and senior engineer officers. At almost the beginning of his service, we find the young officer mentioned for "the great assistance rendered in repairing the boilers, often without drawing the fires." Captain Collins, of the *Wachusett,* with justice, but extreme conservatism, gave it as his official opinion that Melville was "a valuable officer to the Government." The captain of the *Penobscot* believed him to be "capable of building and running the engines of any of our men-of-war." On the *Lancaster,* it was said officially that he "exhibited an amount of mechanical ability, energy,

and engineering skill rarely found." And the fleet engineer of the United States Asiatic Squadron, later the honored Engineer-in-Chief Loring, U. S. N., wrote officially, as to his service on the *Tennessee,* that experience with Melville but confirmed "the high reputation this gentleman has throughout the service for professional skill, executive ability, energy and zeal. In all these qualities, as well as in those other essential ones that go to make up true manhood, it is no disparagement to his fellows to say that I believe he has not his superior in his corps"—an opinion which De Long, in later days, with the *Jeannette* beset in the pack, found occasion also officially to write.

It might well be assumed that Melville's tireless and intrepid spirit would not rest easily beneath "the rust of a long peace." With no foes of the flag to fight, he turned at the first opportunity to a strife perhaps more perilous,—the war with the "White North."

In 1870, Captain Hall, the American explorer, made his last voyage to the Arctic, reaching the hitherto inaccessible polar ocean. His ship, the *Polaris,* was then forced to the Greenland shore and there, in Thank God Harbor, Hall died. Subsequently, in returning southward, the *Polaris* was almost crushed in the ice and a part of her crew fled in the darkness to the floe, on which they drifted 1,300 miles during five terrible months, and were rescued finally, off Labrador, in April, 1873, by the sealing steamer *Tigress.*

When news of this reached the United States, the *Tigress* was chartered at once to return for the remnant of the expedition. Melville volunteered as her engineer officer; and, although the machinery of the vessel was worn and old, he succeeded in driving her as far north as Littleton Island, the winter quarters of Dr. Kane, in a little over thirty days. The camp of the *Polaris* party was found, but they had deserted their stranded ship and gone southward, having been rescued finally by a whaler at Cape York.

Unaware of their safety, and during the fierce autumnal gales of the Arctic, the little *Tigress* pursued gallantly her

search, "being able to keep the sea under steam," as, on his return, Commander Greer, U. S. N., stated officially, despite "the miserable condition of the boilers and the cheap machinery found aboard," through Melville's "great fertility of resource, combined with thorough practical knowledge."

The ice hunger was fully on him now,—that strange desire, which, for more than three hundred years, has drawn poleward, to "the Ultima Thule of the world," so many of the stern and dauntless spirits of our race, from John Davis, of Sandridge, who in 1588, in his tiny vessels and with his reckless gallantry, made the first great dash toward the Northwest Passage, to Melville, Greeley, Nansen and Peary of our time. And so, when six years later, the next Arctic expedition left the shores of the United States in the *Jeannette*, we find Melville with it, an eager and daring volunteer.

The *Jeannette*, a bark-rigged steamer of 420 tons, was purchased and equipped for polar service through the liberality and patriotism of Mr. James Gordon Bennett, of New York. She was commanded by Lieutenant George W. De Long, U. S. N., serving with whom were Melville, two lieutenants, and Surgeon Ambler, of the United States Navy, and twenty-eight others comprising the scientific staff and crew. Melville's detail was not obtained without effort, De Long noting that "the Department was very reluctant to attach him to the *Jeannette* from a sense of the extreme difficulty of supplying his place in his absence."

Prior to this voyage, no polar expedition had ever passed through Behring's Strait, although high northing had been made by British ships, sent by that route, years before, in the hope that they might meet Sir John Franklin's squadron. In facing so much that was unknown, the expedition was regarded as but preliminary and experimental in seas which no keel had plowed.

Leaving San Francisco, Cal., on July 8, 1879, the ice was met early in September a short distance northwest of the straits. The vessel never, thereafter, escaped from the floe which held her. She drifted almost steadily to the westward

for nearly two years, and, in the end, was crushed and sunk, June 12, 1881, in 77 degrees 15 minutes N., 155 degrees E., leaving her crew shelterless on the ice floe in mid-ocean.

During the long and dreary months of forced inaction in the ice, De Long learned to know the high qualities of his engineer officer, as the pages of his *Ice Journal* show so often. "Melville," he notes, "is more and more a treasure every day. He is not only without a superior as an engineer, but he is bright and cheerful to an extraordinary extent. He sings well, is always contented, and brightens everybody by his presence alone. He is always self-helpful and reliant, never worries about the future, is ready for any emergency, has a cheerful word for everybody night and morning, and is, in fine, a tower of strength in himself."

As he wrote his manly words, De Long had no thought of the bitter fate in store for him. He little knew that the genial and stout-hearted man whom he praised would yet, through storm and danger, seek and find him in death. That the strong, sweet voice, whose songs cheered so often that hapless company, when beset in the ice and on the long retreat, would yet chant, in sorrow, his requiem above the lonely tomb on the Lena Delta.

Life in the floe was not wholly one of ease and peace. The ice was restless and the ship met many a jar in her long drift. She had been but four months in the pack when she was "nipped" and strained almost fatally, the water coming in rapidly from forward and freezing, as it entered, in a temperature of 29 degrees F. below. The deck pumps were manned and seemed, for the time, to hold their own; the boiler was filled hastily with snow, slush, and ice from the bilges; and the furnaces were fired long before this was completed. At last, steam drove effective pumps, and the *Jeannette*, for a space, was saved by its power. De Long speaks of Melville's "indomitable energy" in this crisis, and, later, of "Melville who will not sleep or rest," but who, when he does retire, "lies awake, planning some new means of pump-

ing a ship by steam which will be more economical than the main boilers."

The latter question was of vital importance. With a limited coal supply, the large expenditure for pumping meant, if not lessened, abandonment of the ship and facing the perils of the moving pack. The commander of the expedition refers often to Melville's success in devising one mechanism after another to accomplish this; and, in view of his words, General Greeley's comment seems but just that the *Jeannette* "avoided foundering only through the energy and skill of Melville."

With little but the ceaseless pumping to disturb the "unbroken monotone" of life within the pack, the ship drifted for nearly eighteen months from the time she sprung a leak, until, on May 16, 1881, an unknown island was sighted. Eight days thereafter another loomed up, and, although the drift was rapid and the floe was in a swirl, De Long sent, to unfurl the flag on this new land, a small force with Melville in command.

The journey was hazardous in the extreme. There were lanes of water to be ferried over, roads to cut, and chasms to fill that the dog-team might cross. The temperature was many degrees below freezing, the clothes of the party were wet, and their hands frost bitten. Undeterred, Melville pushed on for three days, when the island rose before him; but between them surged a wild stream of ice blocks in constant motion and tumult. "Great bodies of ice were incessantly fleeing, it seemed, from the mad pursuit of those behind; now hurling themselves on top and now borne down and buried by others. And it was through this chaos that we must force our way to the island."

Abandoning their light boat, with much of their provisions, Melville and his men made a dash for the shore; dragging the dogs which, from fear, refused to follow; jumping the sled from one to the other of the rolling floe bits; and reaching, at last, wholly exhausted, the coveted land. Giving his flag to the breeze and taking possession of the new territory—as Henrietta Island—in the name of the United States, Melville

made a running survey of the land, with the bearings of the principal promontories; and then, after a short rest, the expedition began its return, during a bitter storm and reaching the ship after five days' absence. Of Melville's reckless gallantry in this, De Long justly says: "If this persitence in landing upon this island, in spite of the superhuman difficulties he encountered, is not reckoned a brave and meritorious action, it will not be from any failure on my part to make it known."

In the early morning of June 12, 1881, the *Jeannette* met her fate. Thirty-six hours previously, the motion of the ice had become violent; later, the floe split and the ship righted; then it closed in again and she began slowly to fill. At 4 A. M., she sank suddenly. The yard-arms were stripped and broken upward, parallel with the masts, and so, like a great, gaunt skeleton clapping its hands above its head, she plunged out of sight. The situation was one to chill the stoutest heart, —out on the split and moving floe in mid-ocean; 500 miles in a direct line from the Lena Delta, the nearest hope of succor; 150 miles from the nearest known land, the desolate New Siberian Islands; with some sick men; and able to transport but a limited supply of food. Of a portion of the retreat from this point, General Greeley says:

"Lieutenant Danenhower had long been disabled, Lieutenant Chipp and three men were sick, and, under these conditions, De Long retreated south to the New Siberian Islands, 150 miles distant. In this fearful journey, Melville was full of energy and expedients."

Forty-one days after the march began, a hitherto undiscovered land, named, by De Long, Bennett Island, was reached. The journey, thus far, had been entirely by sledging and had been attended by herculean toil. Throughout it, roads had to be cut, bridges built over narrow leads, and rafting resorted to where bridges would not serve. The force was, in addition, too small to haul more than one sled at a time, and, therefore, to advance all of the baggage, the course must be traversed, to and fro, thirteen times.

Melville led the working force during the greater part of

the distance to Bennett Island, acting as a lead horse in setting the pace, or tugging in the harness and putting his shoulder to the sled, when need arose. De Long writes of this: "There is no work in the world harder than this sledging; and, with my two line officers constantly on the sick list, I have much on my hands. In Melville I have a strong support as well as a substitute for them, and as long as he remains as he is, strong and well, I shall get along all right."

A stay of nine days on Bennett Island recruited the exhausted men, and, as the Arctic winter set in, they started southward again. There were many water leads in sight, and De Long embarked his crew in the three boats, commanded respectively by himself, Chipp and Melville. The latter's orders were explicitly "take command of the whale boat;" and this was emphasized thus: "Every person under my (De Long's) command at this time, who may be embarked in that (whale) boat at any time, is under your (Melville's) charge and subject to your orders."

The journey, while now less fatiguing, was still attended by great toil and suffering. Sailing, for any distance, was impossible. Again and again the shifting ice would block a lead, and the boats must be discharged, hauled up, and dragged, perhaps half a mile, to their relaunching. While in the boats, the men were cramped, cold, and wet; their scanty clothing was in rags; and they had no covering for hands or feet. Once progress was totally stopped and the "Ten Day Camp" was made upon the floe. Several times they touched at islands, but lingered briefly, for food was failing and delay meant death.

At last, after five weeks of this, we find the three commanders, on the evening of September 11, at the edge of the ice, with the open sea before them, and the Delta but ninety miles away. They communed together long and earnestly. Help seemed near and hope surged high. And yet, all unaware, there clung, about these stern and gallant men, the shadow of death. They had come to the parting of the ways.

The little squadron was in motion with the dawn, Melville

attending the sheet and retaining then, as always, the command of his boat. The wind and sea arose, and by evening it was blowing a living gale, before which the boats ran. The swifter whale boat forged ahead and could not keep the position, with regard to him, that was indicated by De Long. With great seas following and threatening to comb over on her, she must either increase her speed, heave to, or be swamped. De Long, seeing her peril, waved her onward; and, with that last signal in the misty twilight, Melville and he parted for all time.

"Looking backward," later on, Melville writes of Chipp's boat: "Once more she appeared; an immense sea enveloped her; she broached to. I could see a man striving to free the sail; she plunged again from view; and, though wave after wave rose and fell, I saw nothing but the seething white caps of the cold, dark sea." Chipp and his men had gone to seamen's graves.

On the whale boat there were eager eyes watching to sight the stream of running ice in which she might find shelter. For her to outlive the gale it was, as Melville tersely puts it, "A case of 'Night or Blücher—ice or heave to!" There was no ice, and, at last, the boat, with danger and difficulty, was put about and lay, head to wind, held by a rude sea anchor which had been hastily made. In suffering and peril, her hapless crew pitched and rolled until the next afternoon, when they were again under way, entering, after many hours at sea, one of the mouths of the Lena river, and reaching, after many privations, the Russian village of Geeomovialocke. In about 100 days and transporting 290 pounds per man, they had completed an unparalleled retreat, aggregating, in its many windings, over 2,200 miles.

Their condition was pitiable. Feeble, ragged, and starving, with frozen limbs, painful as though scalded from knees to toes, they clambered ashore, Melville himself being unable to stand for fifteen days. The village was poor and desolate. The only food was geese and fish, both well advanced toward decay, and there was little even of these. With enfeebled

men on such a diet and without anti-scorbutics, scurvy became a present danger, and departure inland a vital necessity. It was, however, only after a lapse of three weeks that it became possible to send a native courier through to Bulun, the nearest official Russian settlement. Two weeks later, he returned bringing some supplies and what seemed to Melville a message from the dead—a dispatch from two seamen of De Long's crew.

Their boat had landed September 17 and they had followed southward the barren shores of the Lena, retarded by snow, young ice, and the sick. De Long and Ambler, to their everlasting honor, had held fast to their feeble and dying men, and, when game and food failed, had sent two of the party in advance to seek relief. From these the message came. Melville set out, at once and alone, for Bulun, and there met the two seamen. He was still far from strong. De Long and his men were dead. Although he did not know this to be so, his judgment told him that, after such a delay, a search would be fruitless in saving life. And yet, "it was duty." Like the heroic Richard Chancellor, of early Arctic history, his noble rule was "either to bring that to pass which was intended, or else to die the death!" An eloquent writer has said of Melville's unflinching fortitude and fidelity at this time:

"To the southward, toward Yakutsk, whither he had ordered Danenhower to proceed, a warmer sun was shining, and there lay the great Russian road leading to the borders of civilization; to the railway, to the blue sea, to home! Few men, after passing through all the perils that Melville had survived, would have thrown away that seeming last chance of personal safety; but he, being cast in rare, heroic mould, turned his back upon the means of saving himself and set his rugged face to the northward, forcing his way into the darkness and the awful silence of the Arctic winter like one deliberately invading the dominions of death."*

And so, starting out with two natives, two dog-teams, and five days' food, he began a search which extended through

* Bennett's "The Steam Navy of the United States."

twenty-three days and over a thousand miles of travel, in the deadly cold of the Arctic winter, and at a time when there were but two hours of daylight in the twenty-four, and through the fierce gales which then swept the treeless *tundras*.

It is not possible to detail fully herein the peril and the suffering of that search. Melville early became disabled from the cold, his feet and limbs swelling, blistering, and the skin cracking. While he stopped occasionally at settlements, food was scarce there and it failed often in the field, necessitating the use of reindeer bones, fish heads, and other refuse, gathered on the march. The faces of the natives became blistered and sore, the dogs were worn to mere skeletons, and even Melville himself grew careless as to whether he lived or died—and still he pressed on! Inured to the cold as they were, his men flagged often in their march; but by entreaty, by argument, by intimidation with the sight and sound of fire arms, the half frozen, helpless, but dauntless leader, kept them moving. Once they mutinied outright, saying:

"We have no food. We cannot go. We shall die!"

The only answer was:

"I will go on. We shall eat the dogs first; and, after that, I will, if necessaary, eat you Yakutsk; but I will go on."

And the mutiny was crushed.

The journey extended to the shores of the Arctic ocean where he found the instruments and records *cached* by De Long, whose track he followed for some distance inland, until, misled by the chart, he lost the trail. Then, exposed to a pitiless storm, attacked by an enervating dysentery, and worn to a shadow from cold, hunger, and slow starvation, he returned to Bulun.

With the earliest spring, despite fierce gales, Melville was again in the field, leading then a well equipped party. On March 23, 1882, he found his dead shipmates,—De Long lying on the river bank, Ambler with him, and the rest gathered near. Apparently Ambler had been the last to die. He lay with a loaded pistol in his hand, as if defending the dead be-

fore him, keeping "his lone watch to the last—on duty, on guard, under arms."

Of him,—a Virginia gentleman of the old and courtly school—Melville spoke always with deep tenderness, for the ties between them were warm and brotherly, born of regard, each for the other's fortitude, fidelity and honor. An incident which shows the manly traits of both may well be given here. The dark horrors of the Franklin retreat, which were revealed when the seekers, years after, found human bones in the camp kettles of that hapless company, had been discussed between them and powders of cyanide of potassium had been prepared on the *Jeannette* to be carried by both, that they might end their lives, if, in the famine of a long retreat, such a temptation should ever come to them.

Again, when near the ending of that last march, De Long broke down—physically, but never mentally,—and, in noble self-sacrifice, asked that his men leave him and go on, Ambler's stern answer was: "If you give up, I take command! And no one shall leave you while I live." They were knightly to the last, these wan shadows of living men.

There remained but to bury the dead. If left on the lowland the coming floods would sweep them to the ocean. In the foot hills, miles away to the southward, there was a rock, "a bold promontory, with a perpendicular face, overlooking the frozen polar sea," and, on its summit, Melville built a tomb of massive timber capped with a heavy cross. Then, turning tenderly the dead faces "towards the East and the sun rising," as he writes, "in sight of the spot where they fell, the scene of their suffering and heroic endeavor, where the everlasting snows would be their winding sheet, and the fierce polar blasts would wail their wild dirge through all time,—there we buried them, and surely heroes never found fitter resting place."

It was done. The working party turned to go, but Melville lingered. He was worn physically, brain weary and heart sick; and, for a moment, the reaction came. The stern eyes that the snow glare could not blind grew dim, as memory,

in full tide, surged backward through the years. He had been the singer of the *Jeannette,* and, leaning on the tomb, there came from him, half unconsciously and yet in noble pathos, his last song to the dead:

We thought, as we hollowed the narrow bed
And smoothed down the lonely pillow,
That the foe and the stranger would tread o'er their heads,
And we—far away on the billow.

Lightly they'll speak of the spirits who're gone
And, o'er their cold ashes, upbraid them;
But little they'll reck, if we let them sleep on
In the grave where a comrade hath laid them.

Slowly and sadly we laid them down
From the field of their fame, fresh and gory,
We curved not a line, we raised not a stone,
But we left them alone with their glory.

The mourner, alone there on the *tundra,* at the bitter ending of his heroic *Odyssey* of wandering in the North, was proving again, all unconsciously in his last song, a truth as old as time:

The bravest are the tenderest,
The loving are the daring.

His words, too, rang truer than he knew, for it was not so long after that the dead commander was "upbraided," that his conduct of the expedition was questioned unjustly, and that Melville appeared before a Congressional commission, his defender in death as he had been his "tower of strength" when ice beset.

De Long and his silent company have been borne since then across the sea and sleep in peace with the dust of their kin. Without detraction from that dead leader's fame, in his masterly retreat and in his intrepid march to death, it may be said that Melville's name glows brilliantly on many a page of the history of that expedition, inscribed there often by De Long's unselfish hand.

Glance at the record of the engineer officer! In the begin-

ning, the *Jeannette* avoided foundering only through his energy and skill; and she would have been abandoned long before she sank, but for his inventive genius and resource. He was the first, through many dangers, to unfurl the flag on unknown Arctic land. He led the working force during the greater part of the fearful march to Bennett Island. He commanded one of the three boats in the subsequent retreat; one of the two which reached the land, and was the only commander to bring his crew through safely.

While yet enfeebled, he made a search, without parallel in Arctic history, for De Long, through wild storms and deadly cold in the sunless day of the Arctic winter. In earliest spring time, amid fierce gales, he found and buried his dead comrades. He recovered and brought back, while ill and partly frozen, all the records of the expedition. And, finally, before turning his face homeward, he made a last search for Chipp and his men, outlining with his party, the entire coast of the Delta and entering the mouths of all its streams. Surely, the force of duty, of honor and of friendship could no farther go than this.

Melville's achievement, in this expedition, retreat and search, met cordial acknowledgment from competent authorities. The Hon. William E. Chandler, Secretary of the United States Navy, referred to it as "untiring and intrepid." In 1890, the Hon. James R. Soley, Assistant Secretary of the Navy, delivered an address at the unveiling of the *Jeannette* monument at the United States Naval Academy at Annapolis, Maryland. While stating that "it is not to the living, but to the dead, that this day is consecrated," the speaker said:

"First of all come his (De Long's) two right-hand men, the executive officer and the engineer, of whom he writes: 'Aided by Chipp and Melville, whose superiors the Navy cannot show, with their untiring energy, splendid judgment and fertility of device, I am confident of doing all that man can do, to carry the expedition to a successful termination.'"

And again of the search, he said:

"There is more yet to be told of this expedition, but we

may not dwell upon it here. * * * How Melville heard from Noros and Nindemann of his captain's cruel plight, and how, before he had regained his strength, he braved alone the Arctic storms and cold and hunger, traversing the Delta down to the very coast, freely putting his life again at hazard in the faint hope that he might bring relief; how a second time he scoured the plains in March and found the bodies, and, laying them in the earth to rest, built over them the cairn and cross, in whose likeness this stone has been erected here."

The naval Court of Inquiry which, in 1883, investigated all facts as to the *Jeannette* expedition, after stating in its report that Melville, in his search, nearly sacrificed "his life from hunger and cold," said that especial commendation was due him "for his zeal, energy, and professional aptitude, which elicited high encomiums from his commander."

Brigadier General A. W. Greeley, U. S. A., the commander of the Arctic expedition to Lady Franklin Bay, in his work "Explorers and Travelers," makes mention of Melville's "indomitable energy and great professional skill" in getting under control the leak in the crushed *Jeannette*; of how, "at the risk of his life, through the moving pack," he landed on Henrietta Island; and of how, "exhausting all practicable means," he pressed the search for De Long.

As to the retreat over the pack in which Melville played so important a part, this eminent Arctic authority says:

"Lieutenant Danenhower being disabled and Lieutenant Chipp sick, De Long's main dependence was in his chief engineer, Melville, who was well, strong, energetic and fertile in expedients. * * * It is unnecessary to dwell on the dangers and hardships which this unprecedented journey entailed on the members of this party, which were met with fortitude, courage and energy that made its successful issue one of the most notable efforts in the history of man, overcoming obstacles almost insurmountable."

In reporting favorably a bill giving Melville promotion for his heroic service, the naval committee of the United States House of Representatives of the Fiftieth Congress said:

"The qualities, as officer and man, displayed by Chief Engineer Melville throughout the cruise of the ill-fated *Jeannette*, and especially during the retreat of her crew over the ice and waves of the Polar sea, to the desolate shores of Northern Siberia, and more strikingly during the search for the remains of Lieutenant Commander De Long and his party during the perils and privations of an Arctic winter, have invoked the admiration of the world."

In the Fifty-first Congress, the Senate naval committee reported on this bill as follows: "In view of such a record, the committee feel that the proposed promotion, so long after the deeds which it is intended to recompense, is an act of tardy justice and hardly adequate."

The bill, as passed finally, stated that the promotion was given as "a recognition of his (Melville's) meritorious services in successfully directing the party under his command after the wreck of the Arctic exploring steamer *Jeannette* and of his persistent efforts, through dangers and hardships, to find and assist his commanding officer and other members of the expedition before he himself was out of peril."

Melville reached home on September 13, 1882, just one year after the parting of the boats in the gale, and, for him, a year of danger and of strenuous toil. On May 1, 1884, we find him northward bound again, this time with the squadron dispatched to the relief of the Lady Franklin Bay expedition. This expedition had been in the Arctic since August 11, 1881, and, having no means of transportation by sea, was to have been relieved, at a predetermined time, by vessels from the United States. The relief ships not appearing in due season, the expedition began its retreat, which was effected, in health with records and all essentials excepting full food supplies, to Cape Sabine in October, 1883. There, hemmed in and daily expecting succor, Greeley made what proved to be the death camp of many of his men, and there the feeble and starving remnant of his heroic command was rescued, June 22, 1884, by the United States ships *Thetis* and *Bear*.

His expedition "failed not in aught entrusted to it"

and "perished through others;" for, as Melville states, the United States Government's efforts for relief, made in due season, "were thwarted by carelessness, incompetency or inexperience." On September 14, 1883, while Greeley's men were still in good condition, Melville wrote the Navy Department, volunteering, if landed at Cape York, to lead a party to Littleton Island, to communicate with Greeley, and, if the latter's men were sufficiently strong, to conduct them to the base of supplies at Cape York.

This proposition was rejected by a naval board as impracticable, mainly on the ground of the lateness of the season, although whalemen had been known to cruise as far north as Cape York up to October 20. General Greeley himself said afterward, as to this: "I have already officially concurred in the views of Chief Engineer Melville and the opinion of the sealers of Newfoundland that our relief was practicable during the autumn of 1883."

With the rejection of Melville's project, which his experience and courage would, doubtless, have made successful, the ill-starred expedition was left to its fate for another winter, another spring, and well into another summer. When, finally, a relief squadron was organized, Melville volunteered; his knowledge of the needs of such an expedition was utilized largely; he was among the first to reach the dying men at Cape Sabine; and, with this mission of mercy, there closed fitly his Arctic service, and, in effect, his service afloat.

Admiral Melville acted as Engineer-in-Chief of the Navy from 1887 to 1903, during a period of sixteen and a half years, which covered the creation of the "New Navy" and the War with Spain. This service, in its duration, and, indeed, in many respects in its achievement, has never been equalled by any Chief of Bureau in the history of the Navy Department. Throughout these years, machinery aggregating a million and a quarter horsepower was designed under his masterful supervision for about 120 vessels, 24 being battleships and 41 armored ships of all classes.

His tenure of office began in characteristic fashion. The

rebuilding of the fleet had been started and four vessels, designed by United States naval engineers and constructors, had been completed by contract in the yards of John Roach, of Chester, Pa. These ships were reported upon adversely by various naval boards, somewhat unjustly it appears, since two at least of them are afloat and in service today. Influenced, however, by these criticisms, the Navy Department purchased abroad the plans of several other vessels for further additions to the fleet. In 1887, the Secretary of the Navy resolved to change the foreign policy which these purchases indicated; and he instructed Chief Engineer Melville to prepare, at Philadelphia, Pa., special designs for the machinery of two 1,700-ton gun boats, and original drawings for that of a 4,100-ton cruiser. When completed, those of the cruiser were accepted by the Union Iron Works, of San Francisco, Cal., which company guaranteed their success, preferring them to foreign designs offered as alternatives. From them the machinery of the cruiser *San Francisco*, a highly successful ship, was built; and there was ended thus the régime of imported plans, those of the Bureau of Engineering having been accepted, as a whole, without question by contractors in all subsequent work. The energy with which, in a few brief weeks, this task was done, with its wholly successful outcome, met due recognition; and, on August 9, 1887, shortly after its completion, Melville was nominated as Engineer-in-chief of the Navy, with the relative rank of commodore. Later, through the provisions of the Personnel Law, he was advanced to the grade of Rear Admiral. Melville's years as Engineer-in-chief were marked from the first by the same wise daring which was the keynote of his whole strenuous life. Its essence was supreme courage joined with the prudence which came to him from his Scotch forebears. From the beginning, in his annual reports, he spoke out boldly, giving without reserve the facts as to naval conditions, as he knew them. This free speech antagonized the conservatism of officers of the old school, and complaint was made to President Cleveland, who—himself a man abrupt

as he was strong and frank—read the report and said simply: "We need more such men."

And so, in the administration of the affairs of the Bureau of Steam Engineering during this period, there was always progressiveness of the right kind. Its Chief was eager in his ceaseless search for all that was best in late developments, but he was never attracted by mere novelty only. Strong as he was to make advances on lines from which an engineer of less courage would have shrunk, he had still the mental poise and strength to hold back when his judgment told him that some suggested improvement was still in the experimental stage.

As soon as he entered office, he virtually revolutionized launch machinery, fitting water-tube boilers and other improvements. Then he overthrew ruthlessly the time-honored tradition that the engines of a war vessel should be horizontal that they might be below the water line. The machinery of the first *Maine* of tragic memory marked his departure to the type of the vertical engines of short stroke and high speed which serve so efficiently today. In boilers, he met from the first, with the utmost skill and courage, a long and bitter struggle. While the cylindrical fire-tube boiler remained in service, the demand for the reduction of machinery weights, coupled with the extended use of forced draft, tempted foreign designers to cut boiler weights too low, with disastrous results in one great navy. Melville refused to be moved by this demand. He held resolutely to a policy of efficiency with full safety, declined to so reduce weights as to make undue forcing necessary, and thus our Navy was free from boiler failures.

When the day of the Scotch boiler was ended, Melville became the pioneer in the world's navies in the introduction of boilers of the water-tube type in large vessels. In designing the machinery for the monitor *Monterey*, the limitations were such as to make an installation of cylindrical boilers impracticable. He proceeded at once to secure a water-tube boiler which would meet the conditions. Competitive tests were made, and the ship was eventually fitted with coil boilers for

three-fourths of her power. That this step was a bold one at that period is shown by the fact that one great shipbuilding firm, which was the lowest bidder, refused to guarantee the power required by the specifications, except as to the two Scotch boilers which formed one-fourth of the ship's equipment. The success of the *Monterey* would have tempted most engineers, with Melville's opportunities, to immediate action on a wide scale. He felt, however, that sufficient experience with the water-tube boiler had not yet been gained, and for that he was content to wait. Meanwhile, in some foreign navies, although they began later than he, the introduction of complex boilers of this type was pressed with such feverish eagerness that there were many errors, great waste of money, and much bitter controversy.

Admiral Melville was always an advocate of high speed in war vessels. To him, speed meant position, the "weather gage" of the old days. In forceful words, he said: "Speed, and particularly its control, form the cardinal elements of tactics," and, in pursuance of that belief, he designed for the United States Navy the machinery for two vessels which, for their size, were the swiftest in the world in their day, the triple-screw cruisers *Columbia* and *Minneapolis*, whose record remained unequalled for nearly a decade. These ships, which were perhaps the greatest triumphs of his professional career, gave marked evidence of both his judgment and his daring. Although urgently advised by many professional friends not to attempt so radical an experiment, he went calmly on—to full success in the end. The engineering problem, as to these ships, was difficult of solution. A hull of 7,375 tons displacement was to be driven at a speed of 22 knots on an indicated horsepower of 21,000. With twin screws this would have meant a thrust of 10,500 horsepower on a single shaft. While the Bureau of Steam Engineering had, from past experience, full confidence in the ability of the great works at Bethlehem, Pa.—under the conduct of that eminent engineer, John Fritz—to produce steel forgings of the highest character, still, up to that date, no shafting for the transmission of this great power had been

made in the United States. Again, a design including twin screws would have necessitated not only massive shafting, for that day, but large parts of the machinery throughout, if with single engines of 10,500 horsepower on each shaft.

A further consideration was that of securing economy at moderate speed, when full speed was so high; and it seemed that this would be better attained by running engines of 7,000 horsepower at, say, 2,000, than those of 10,500 horsepower at a corresponding speed. The only further alternative was a twin-screw design with two engines of 5,125 horsepower each on the same shaft—a method which involved much multiplication of parts, as well as disconnecting gear for one engine on each shaft, and special provision for the wearing down of the shaft bearings of the after engines.

The French experiments on the *Carpe* and the trials of the *Tripoli* had left no doubt as to the success of triple screws for those powers; and the French had shown their faith in them still further by the preliminary designs of the *Dupuy de Lôme*. In view of this and of the consideration above set forth, the engineer-in-chief decided to use triple screws in these vessels. Their success, in contract trials and in long endurance runs, amply vindicated his judgment.

On other occasions, Admiral Melville argued effectively for high speed in our warships. When, in 1898, the proposals for the new battleship *Maine* and her sisters, the *Missouri* and *Ohio*, were in course of preparation, he stood alone in his demand for 18-knot, as against 16-knot, ships, and, single-handed, he began the fight against the indifference in some quarters and the active opposition which existed in others. His forceful protest was taken up editorially by the "New York Tribune," and soon found nation-wide approval. If the Admiral had not won, as he did, we should have been, for three years longer, behind the great navies of the world in battleship speeds. Again, in 1902, he made a similar and most vigorous effort when, as a member of the Board on Construction, he fought for the highest practicable speed for the armored cruisers *Washington* and *Tennessee*.

Lack of space forbids any attempt to review fully Melville's career as Engineer-in-chief. It may be said, however, that such a review would be largely but a record of pioneer progress in the science of marine engineering. He was the first to determine by continuous tests the actual coal consumption of naval boilers, with results which were somewhat startling to those who had been "estimating"—by guess—the steaming endurance of our ships. Again, he was the first to use the "standardized-screw" method of finding the trial speeds of ships, a method which has withstood the test of time as at once the simplest, the cheapest, the most accurate, and the fairest to the contractor and the Government. During his administration there were conducted the most extended series of tests to ascertain the relative values of coal and liquid fuel which have ever taken place. The report of these tests has become a world-classic on this subject. As early as 1903, he recommended officially that a ship, preferably of the scout or fast-cruiser class, be fitted with steam turbines to try out the value of the latter for naval purposes. He urged for years, and with final success, the establishment of the Experiment Station and the Post-Graduate School of Marine Engineering at Annapolis. The latter especially lay near his heart, for, like the Scots from whom he came, he had the love of learning for learning's sake. His foresight as a naval engineer, marked in many minor instances, was especially brilliant in four most noteworthy acts: the use of triple screws, the early adoption of the water-tube boiler, the tests of fuel oil whose great future he predicted, and finally the anticipation in design by him of the *Dreadnoughts* of today. For years, he had advocated what he called an "all big-gun, one-caliber ship," and, in 1899, he submitted to the Board of Construction a sketch plan of the battery of such a ship, carrying ten 12-inch rifles and having a speed of twenty knots.

In the writer's view, Admiral Melville's greatest service, both to the Navy and to his profession, lay in the noble work which he did in widening the field and in increasing the prestige of the engineer. In these pleasing days of naval harmony, one

can look backward, dispassionately and with no bitter memories, on that long struggle of the old Engineer Corps for full recognition in our Navy. Viewed thus, that struggle seems now natural enough. On the one side, there were arrayed men trained as sailors, intensely proud of the traditions of a most brilliant past; on the other, men skilled in a new science, which, with their knowledge, they could readily foresee was fated to drive the sailing warship from the seas. The *fuit Ilium* of the old days and the old Navy had been sounded, but the Line of that time were reluctant to heed the warning.

The struggle between the two forces, thus honestly and fairly opposed, was long and severe. In the end, the entire engineering profession of this country was united in demanding recognition for their brethren of the Navy. The League of Associated Engineers, comprising ten thousand professional men of all branches, was organized for this purpose and contributed liberally to forward it. The primary influence in stirring these men and many others to action, was Melville's towering personality. With his heroic record and his masterful genius, he seemed to them like a Viking of old days, whose lion-like leadership they were proud to follow. In the happy issue of it all, when peace with honor came to the officers of both branches, Melville was the first to welcome it, eager always, in his great-heartedness, to sweep away dark memories and heal the wounds of that long strife. Ericsson's prophetic dictum of years ago, was that "modern naval warfare must henceforth be largely a matter of engineering," which is an axiom now. And so, the world's navies and the nations whose far-flung lines of battle and defence they are, owe Admiral Melville a deep debt for his long struggle for, and his advancement of, the cause of engineering on the seas.

Melville was remarkable, not only for his lucid thought but for his abundant and apt vocabulary in its expression. In speech as in action, he sought the objective instinctively, and then, like an arrow to its mark, his powerful intellect made a straight course, undeflected by the irrelevancies, the side issues, which trouble feebler minds. In April, 1884, the "Jeannette

Inquiry" was held by a sub-committee of the Committee on Naval Affairs of the House of Representatives. In this investigation, the heirs of De Long were, with Melville, represented by counsel, to whom was opposed the counsel for the petitioner. Melville's testimony—or rather argument—before the committee excited comment for its clearness, its unbroken flow, and its sustained power during long hours. After referring to it as "vivid and beautiful," the opposing counsel stated:

"I have this to say with reference to Melville, that he is mentally so constructed as to be an extraordinary and remarkable man; and, in that wonderful oration which he delivered here, under oath, from half-past 10 in the morning until 6 at night, I have never witnessed, in my experience, more physical and mental endurance."

These words—from, for the time, a formal foe—indicate partially Melville's power in written and spoken thought. He had, in high degree, the faculty, so essential in literature and art, of minute observation as to, and persistent memory of, details; with further, a vocabulary unusual in its extent, and instant and fluent in its application. Combine these with a wide reading which crops out here and there in simile and quotation, with an imagery drawn from scenes which not all have viewed, and a toil and suffering which few have met, and with the further fact that his honor and that of the dead had been questioned; and it will not, then, seem strange that the "oration" ranged from fierce invective to the deep tenderness and refinement of feeling shown herein as to the burial in the Delta, and that the learned counsel found it "vivid and beautiful" and "wonderful" in its "physical and mental endurance."

In his later years, Melville sometimes told, to audiences whom he held in thrall, the story of his work and deeds in the Far North. Doctor St. Clair McKelway writes, in the "Brooklyn Eagle," as to these addresses:

"The 'lion-hearted' Melville! No one who ever heard the veteran Arctic hero describe the retreat from the lost *Jeannette*,

the weary search for the dead DeLong, a tale of peril and of death, of courage and of loyalty, unsurpassed in the annals of any time or country, but would instantly concede that of all men, the Admiral who died yesterday deserves this distinction. Something in his imperious, impetuous personality carried the listener along until all the terrible scenes of disaster, frost and famine stood before him in effective realism, and made an indelible impression."

The writer recalls an instance of this kind, when the Admiral and Joseph H. Choate were to be the speakers of the evening before the Engineers' Club of New York City. Melville told his story to listeners who were spellbound, until midnight came. Then he ended abruptly, and the silence was profound, for there had been the mist of tears in his gallant eyes and his strong voice had broken, when he spoke of the farewell to his dead comrades as they lay beneath the cairn in the Lena Delta. At last, Mr. Choate rose and said simply:

"After that thrilling story, as noble in its devotion and its self-sacrifice as it is tragic in all else, no words of mine would have fitting place here."

There was one reason for Melville's eloquent and tender language when he chose, to which he very infrequently, and then half shyly, referred. He had been trained in youth by a godly old Scotch mother, a rigid Covenanter, in many a chapter and verse of that "well of English undefiled," King James' version of the Bible. In word and phrase and paraphrase, the lessons of those early days showed themselves unforgotten in his later years.

In his homeward journey from the Lena Delta, Melville was granted a private audience by the Czar and Czarina of Russia at the palace of Peterhoff, near St. Petersburg. In New York, he was received by the mayor and council and given the freedom of the city. Later, a public reception was tendered him in the governor's room at the City Hall and a public dinner. He was given a public reception by the mayor and councils of the city of Philadelphia and a military escort and public reception at Washington. He was elected a member of the National

Geographical Society of the United States, an honorary member of the Royal Swedish Society of Anthropology and Geography, and a member of the Geographical Society of Philadelphia.

In succeeding years, honor after honor flowed in on him, as the world came to know the full measure of the man. By Act of Congress, he received a gold medal and promotion by one grade for his heroic services in the Arctic. The Czar made him a Knight of St. Stanislaus of the First Class. He was elected an honorary member of the Institution of Naval Architects of Great Britain, of the American Society of Mechanical Engineers, the American Society of Civil Engineers, and the Franklin Institute. He was granted the honorary degree of Doctor of Engineering by the Stevens Institute of Technology, of Doctor of Laws by Georgetown University, of Master of Science by Columbia University, and of Doctor of Science by the University of Pennsylvania. He was a Past President of the American Society of Mechanical Engineers, and, at the time of his death, was Commander-in-chief of the Military Order of the Loyal Legion. He sought none of these honors. His services in Polar Seas, his brilliant achievements as an engineer, and, more than all else, his noble qualities as a man, brought them to him. "To valor, heroism, intellect and science, we offer this tribute of respect," were Doctor Weir Mitchell's words, in presenting him for the honorary degree of Doctor of Science to the Provost of the University of Pennsylvania.

Melville's personal characteristics were such as one instinctively expects in the man, the brief outline of whose honorable and honored career has been given herein. He was, first of all, fundamentally frank and straightforward. He had none of the arts of the diplomatist, and spurned all devious ways in working out his ends. Intolerant always of sham and pretence, he met ostentation and evasion with fiery indignation and no measured words. He was rigidly honest and incorruptible. Administering a great office with but a relatively slender salary, and controlling indirectly the character of con-

tracts involving hundreds of millions of dollars, he left that office, after long years of service, a comparatively poor man. Such wealth as he had, came to him in later days. Throughout the whole of his long life, from buoyant youth to weary age, his dominating feeling was an intense loyalty to his country and to the flag he served so well. Instinctively, he seemed to believe himself, naturally and of right, an integral part of the Government he served. Its interests were his; the two were inseparable.

As an officer and as an engineer, the gospel of his life, to the very last, was "Work." But a few days before his death, he was discussing in Washington the designs of machinery for the latest battleships. The officials whom he was consulting were much younger than he, but he kept to their pace until far into the night. Then, they urged him to rest, but he refused. "I may be dead tomorrow, or I may never get a chance to work here again. Let's do this now!" were his words, as typical as they were sadly prophetic. From subordinates he exacted also the full measure of their abilities, but he asked nothing which he did not give himself—work to the limit of his time and strength. In the long retreat from the sunken *Jeannette,* when its hapless crew hauled 290 pounds per man over the shifting floes, it was Melville, always hopeful, always tireless, who led the advance, bending his broad back in the sledge harness, and, to cheer his men, singing the tender old Scotch songs of his long vanished youth.

He was primarily an executive, and, as such, he was a keen judge of character, with a singular faculty of not only attracting young men to his service but of getting from them the best that was in them. To the very ablest among them, they felt it an honor to be on his staff, and gave him devotion in unstinted measure. He repaid this to the full, acknowledging freely and often their aid, and, fearless always of consequences, taking himself the full responsibility for any inadvertent error which they might make. The doctrine of the "Square Deal" was ingrained in him. His dauntless courage in facing, unafraid, any responsibility which his office laid on

him, was a marked mental aid to him in his executive work. When once a given line of action was decided upon, his anxiety ended. The die, for him, was cast. He could carry any load without worry.

In his personal relations, Admiral Melville was a man of broad sympathies and most kindly nature. Throughout his life, his customary salutation to all men was "Brother." As always with the strong who do things, his feelings, for good or ill, were intense. He was imperious and impetuous, aggressive but never vindictive, and severely intolerant of sham, deceit, and incapacity; but, with all this, he was the stanchest and most generous of friends and tender as a woman toward those for whom he cared. John Knox, in speaking of a Melville of his day, as "a man of nature most gentle and most modest" in his friendships, uttered prophetic words as to the Melville of our time. And, too, with all his sturdy independence, the Admiral was, as well, singularly dependent on the affection of those who were near his heart. As the loneliness of age came and friend after friend went beyond the "tideless sea," he clung more closely to those who remained. When his first seizure came, and he felt that he was within death's shadow, he summoned a few of his old staff who were near by, saying simply: "I want to see you once again before I die." Throughout those swiftly passing days of suffering, the man of iron, who had met unafraid the fierce conflict with the White North, seemed to hold as trifles all the honors he had won; he looked only backward to the friendships of long ago and forward, gently and calmly, to his own passing, stirred by what Daniel Webster once called the dominating thought of his later years: "My personal responsibility to Almighty God."

And, now, Melville, the "lion-hearted," sleeps well, by the winding river, in the city that he loved. No Polar blast, no sudden call to arms, will break his quiet rest. With the bowed heads of famous men beside his flag-draped bier, the wailing note of "Taps," and the crash of farewell volleys, earth's last honors came to him. In his life of strenuous toil, his

leadership in great crises, and his supreme self-sacrifice when need came, he answered to the full that world-old cry:

"Give us men from the sunless plain,
By need of work in snow and the rain,
Made strong and brave by familiar pain."

And, when all is said, his most enduring monument will be, not the great deeds of the Admiral by land and sea, but the tender, reverent memories of Melville, the man, which, all unconsciously, he has stirred. Time passes and fame fades, but love and friendship are eternal.

WILLIAM LEDYARD CATHCART.

ON THE BEST DISTRIBUTION OF LOAD AMONG A NUMBER OF SIMILAR STEAM-TURBINE STAGES WORKING IN SERIES.

By E. Buckingham, Associate.

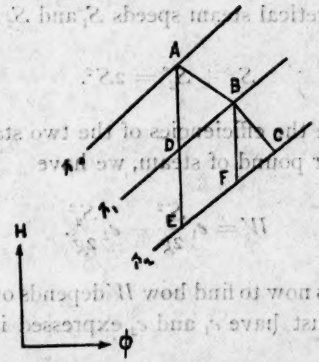
1. STATEMENT OF THE PROBLEM.

Let us consider two impulse stages working in series between a fixed initial steam state p_0 t_0 and a fixed back pressure p_2. The stages shall have the same efficiency curve when the efficiency is plotted in terms of the ratio of r.p.m. to steam speed S, as abscissa. If the pitch diameters are the same we may use the blade speed u instead of the r.p.m. and let the curve be plotted in the usual way with the efficiency e against $\frac{u}{S}$. By the efficiency of a stage is meant here the net work delivered to the shaft inside the bearings, per pound of steam, divided by the work value of the isentropic heat-drop available in the stage from the initial steam state to the final pressure. Windage, bushing leakage, and unused carry-over, but not bearing friction, are included in the wasted fraction $(1 - e)$ which goes into reheat.

If the admission arc to the first stage is kept constant while that to the second is variable by cut-off valves, the intermediate pressure p_1 may be so adjusted as to divide the load in any way we please between the two stages. We will consider the question of what division of the load will give us the maximum output per pound of steam at a given r.p.m.

If the total isentropic heat-drop available is such that an equal division brings each stage to the top of its efficiency curve, then an equal division is evidently the best possible; but if this is not the case, *i. e.*, if the turbine, with fixed initial

conditions and back pressure, is running considerably faster or slower than is required for the conditions just mentioned, it is not at first sight obvious that an equal distribution of the heat-drop, and therefore of the load, will necessarily give the best economy, so that the question is not altogether without interest.



If ABC represents the expansion line on the Mollier diagram, the reheat in the first stage sets the point B off to the right and makes the isentropic heat-drop BF, which is available for the second stage, greater than DE. But if the terminal pressures p_0 and p_2 are not far apart, as when the two stages are elements in a multistage turbine, the isopiestics p_0, p_1 and p_2 will be so nearly parallel that this effect of varying efficiency and reheat in the first stage on the isentropic drop available for the second will be negligible, and we may consider the two stages together as working on a sensibly constant total drop represented by AE.

2. EXPRESSION FOR THE OUTPUT.

To start with, let p_1 be so adjusted by the second-stage cut-off that the same isentropic drop H_0 is avilable for each stage. Let S_0 be the theoretical steam speed corresponding to H_0 so that

$$S_0 = \sqrt{2gJH_0} \qquad (1)$$

Now let p_1 be varied by changing the cut-off. The isentropic heat-drops H_1 and H_2 available in the two stages will now be different, but their sum will remain sensibly constant and we shall have

$$H_1 + H_2 = 2H_o, \tag{2}$$

so that the theoretical steam speeds S_1 and S_2 must satisfy the equation

$$S_1^2 + S_2^2 = 2S_o^2. \tag{3}$$

If e_1 and e_2 are the efficiencies of the two stages and W the whole output per pound of steam, we have

$$W = e_1 \frac{S_1^2}{2g} + e_2 \frac{S_2^2}{2g}. \tag{4}$$

Our problem is now to find how W depends on S_1 and S_2, and to solve it we must have e_1 and e_2 expressed in terms of the steam speeds.

3. THE EFFICIENCY CURVE.

The efficiency of an ideal impulse stage computed from a velocity diagram with no losses except the carry-over, would give a parabola when plotted against $\frac{u}{S}$. Experiment also shows that the actual curve is nearly a parabola, unless $\frac{u}{S}$ is very far from the value which makes e a maximum.

There are various evident reasons why the curve cannot be exactly a parabola and why there is, in fact, no single curve $e = f\left(\frac{u}{S}\right)$ for a given stage. In the first place, if the isentropic heat-drop, and therefore S, is left constant while the rate of revolution and the blade speed change, the windage loss increases as some higher power of u than the first. Hence, even if the efficiency curve can be represented sufficiently well by a parabola with vertical axis for blade speeds up to that which gives the maximum efficiency, the curve

beyond this point will fall faster than the parabola, on account of the rapid increase of the windage.

As a second supposition, let $\frac{u}{S}$ remain constant, but let both u and S be increased by speeding up the turbine and increasing the pressure drop. The losses in both nozzles and blades depend on the absolute value of the steam speed, so that a constant value of $\frac{u}{S}$ cannot possibly mean an exactly constant efficiency. Furthermore, when we increase S in the same ratio as u, so as to keep $\frac{u}{S}$ constant, it makes a difference whether we do it by increasing the initial pressure or by decreasing the back pressure; for the back pressure affects the windage loss, while the initial pressure does not.

Nevertheless, if the stage is working under conditions which are not very different from those under which the efficiency curve was determined, it is sufficiently exact, for practical purposes, to use a single efficiency curve and represent it by the equation of a parabola.

4. THE BEST DIVISION OF LOAD WHEN THE EFFICIENCY CURVE IS A PARABOLA.

Let the vertex of the parabola be at $\frac{u}{S} = a$, $e = b$, so that its equation may be written

$$K\left(\frac{u}{S} - a\right)^2 = b - e, \tag{5}$$

K being the constant which determines the size of the parabola. We then have for the two stages, working on different heat drops, which give the theoretical steam speeds S_1 and S_2,

$$\left.\begin{aligned} e_1 &= b - K\left(\frac{u}{S_1} - a\right)^2 \\ e_2 &= b - K\left(\frac{u}{S_2} - a\right)^2 \end{aligned}\right\} \tag{6}$$

By plotting the efficiency in terms of $\frac{u}{S}$ and setting u the same for both stages, we are evidently assuming a constant pitch diameter, but if the stages are so designed as to have different pitch diameters but the same efficiency curve in terms of $\frac{\text{r.p.m.}}{S}$, all the relations will come out as they do here. The case here assumed is a very common one.

If we now insert the values of e from (6) in equation (4) we get, after a little rearrangement,

$$W = \frac{1}{2g}\left[(b - Ka^2)(S_1^2 + S_2^2) - 2Ku^2 + 2Kua(S_1 + S_2)\right] \quad (7)$$

in which all the terms but the last are constant, $(S_1^2 + S_2^2)$ being constant by equation (3).

To make the output, W, a maximum we must therefore have

$$d(S_1 + S_2) = 0 \text{ or } dS_2 = -dS_1. \quad (8)$$

But by equation (3)

$$d(S_1^2 + S_2^2) = 0 \text{ or } dS_2 = -dS_1\frac{S_1}{S_2}, \quad (9)$$

so that the maximum condition (8) can be satisfied only when $S_1 = S_2$, and the intermediate pressure p_1 is set so as to give each stage one-half of the whole isentropic heat-drop available. The most advantageous setting of the cut-off is therefore that which makes the heat-drop, the efficiency, and the load the same for both stages.

Though this result is not at all surprising, it is nevertheless, not self-evident. It might very well seem possible, for example, that with the turbine running above its most economical speed, *i. e.*, working on too small a heat-drop for the given r.p.m, it would be advantageous to throw most of the heat-drop on one stage and sacrifice the efficiency of the other stage for the sake of the one. The above reasoning shows that this will never be the case if both stages have the same parabolic efficiency curve.

To make the heat-drops equal, the second-stage cut-off

must be set so as to make $\frac{p_1}{p_2}$ slightly greater than $\frac{p_0}{p_1}$. For, whereas with an ideal gas—and very approximately with air—the jet speed depends only on the pressure ratio, the relation is less simple for wet steam. The isentropic heat-drop, $H_{1,2}$, available between any two pressures along a given isentropic line may be represented roughly by an equation of the form

$$H_{1,2} = A \log \frac{p_1}{p_2} + B(p_1 - p_2), \qquad (10)$$

or fairly closely by one of the form

$$H_{1,2} = C \log \frac{p_1}{p_2}(1 + D \log p_1 p_2). \qquad (11)$$

Either of these equations shows that at high absolute values of the pressures a given pressure ratio causes a larger heat-drop and therefore a larger theoretical steam speed than the same pressure ratio taken lower down on the diagram.

5. EXTENSION TO A SERIES OF SIMILAR STAGES.

Let us now consider a considerable number of stages, working in series, and all similar, to the extent of having the same efficiency curve when e is plotted against $\frac{\text{r.p.m.}}{S}$, or against $\frac{u}{S}$ if the pitch diameters are all the same. We also suppose, as before, that this curve, in the part used, is sensibly a parabola. Intermediate cut-off valves are to be fitted so that with given terminal pressures the intermediate pressures may be adjusted at will.

To start with, let the valves be set so that the load is unevenly distributed, the isentropic heat-drop available differing from stage to stage. Let A and B be the two stages—not necessarily adjacent—which have the greatest and least heat-drops. By a proper readjustment of the valves on all the stages except the first, where the admission area is to be constant, it will be possible to adjust the heat-drops in A and B to equality while the drops for all the other stages are un-

changed. To take a definite case, let A be the first stage and B the last. If A and B are to be brought to equality of isentropic drop available, the drop in A must be reduced and that in B increased, and this requires that all the intermediate valves shall be closed a little so as to raise all the pressures enough to keep the heat-drop for each stage except A and B the same as before. The intermediate pressures will not, of course, all have to be raised either by the same amount or in the same ratio, as is seen by consulting equation (11).

The equalization of the drops in A and B increases their total output, for the reasoning of section 4 is not invalidated by the presence of the other stages, if their drops all remain unchanged. The result of our readjustment of pressures is therefore to increase the total output of the turbine per pound of steam, and also to bring the loads on A and B nearer to the mean load on all the stages.

If we go on to the two stages which now have the highest and lowest heat-drops available, we produce another change in the same direction, and by continuing this process of readjustment indefinitely, we gradually approach, as a limit, a condition where all the stages are equally loaded and the total output per pound of steam is a maximum.

The foregoing reasoning is valid on the supposition that the initial and final isopiestics on the Mollier diagram are sensibly parallel, for we have agreed to disregard such small errors as are involved in using a single efficiency curve under all the running conditions met with in the course of the problem. In reality, on account of the fan-shaped arrangement of the isopiestics, a given amount of dissipation and consequent reheat is less wasteful at high than at low temperatures and pressures, and therefore in the earlier than in the later stages of the series. In the case of only two stages, therefore, the maximum output will be obtained by making the isentropic drop a very little greater in the first than in the second; and with a series of similar stages the very best result will require that the heat drops shall not be exactly equal but shall decrease very slighly but uniformly as we go from the feed to the exhaust end.

U. S. S. "FLORIDA."

U. S. S. *FLORIDA*,

DESCRIPTION AND OFFICIAL TRIALS.

By Henderson B. Gregory, Associate.

Battleship No. 30, the *Florida*, is a four-screw Parsons turbine vessel designed for a speed of 20.75 knots, at 21,825 tons displacement and with the main engines developing 28,000 shaft horsepower. She was built at the Navy Yard, Brooklyn, New York, under an Act of Congress approved May 13, 1908, authorizing the construction of two first-class battleships, and which further provided that at least one of the vessels be constructed at a navy yard. The cost, authorized by a subsequent Act of Congress approved March 4, 1911, was $6,400,000.00 exclusive of armor and armament. Her sister ship is the *Utah*, built by the New York Shipbuilding Co., of Camden, New Jersey.

The *Florida* was placed in commission on September 15, 1911, at the Navy Yard, N. Y., and after a brief sojourn on the Atlantic Coast she sailed south, spending the major part of the winter breaking-in at Guantanamo. Late in February she was ordered north to undergo her official trials.

Four trials were required as follows:

(a) A progressive trial over a measured-mile course for standardizing the screws.

(b) A full-speed trial of four hours' duration in the open sea at the highest speed obtainable, with an average air pressure in the ash pits not exceeding 2 inches of water and not over 185 pounds steam pressure above the atmosphere at the M.H.P. turbines. The average speed to be at least 20.75 knots.

(c) An endurance and coal-and-water-consumption trial of twenty-four hours' duration in the open sea, at as nearly as possible a uniform speed of 19 knots, the average not to

fall below that figure. The trial to be conducted as nearly as possible under cruising conditions.

(d) An endurance and coal-and-water-consumption trial at 12 knots under similar conditions to the preceding trial.

The designed water consumption on trials (b), (c) and (d) was not to exceed 15, 16.25 and 26 pounds, respectively, per hour for main engines and auxiliaries per S.H.P. of the main engines. The water and coal on these trials was carefully measured.

The data obtained on the various trials will be found in tabular form following the description of vessel and machinery.

HULL DIMENSIONS.

Length between perpendiculars, feet and inches	510–00
on L.W.L., feet and inches	510–00
over all, feet and inches	521–06
Breadth, extreme, at L.W.L., outside of armor, ft. and ins.	88–02⅝
molded, feet and inches	87–10½
Depth, molded, main deck at side M.S., feet and inches	44–05⅞
Draught to L.W.L., feet and inches	28–6
Displacement corresponding, tons	21,825
Ratio of length to beam	5.781
Displacement per inch at L.W.L. tons of S.W., tons	73.82
Area of midship section, square feet	2,458
L.W.L. plane, square feet	31,056
wetted surface, square feet	56,700
Coefficient of fineness, block	.5837
Length of fireroom space, feet and inches	102–00
engine-room space, feet and inches	60–00

DESCRIPTION OF DECKS, BRIDGES, ETC.

Masts.—There are two masts of the cage or basket type, located at frames 49 and 67, on which are located searchlight platforms, portions of the wireless telegraphy outfit, signal yards, etc.

Bridges.—There are three bridges: steering platform, flying bridge and signal bridge, the latter being the lowest. There are no after bridges.

Forecastle Deck.—This deck extends from the stem to frame 51. On its after part is a deck house containing the admiral's

and captain's quarters. On its level are skid deck beams, in which are stowed all boats except the life boats, which hang from davits aft and the two fifty-foot steamers, which stow on the main deck.

Main Deck.—The main deck extends from stem to stern. Forward of frame 49 is the wardroom country. Abaft this frame, situated amidships, are deck houses containing the crew's galley, officers' galley, butcher shop, bakery, blacksmith shop and foundry and two boat cranes, one on each side at frame 63. The crane towers also serve as additional search-light platforms.

Gun Deck.—The gun deck is also continuous and the full width of the ship. It has within it forward of the casemate the quarters for warrant and for junior officers, the latter being on the starboard side. Within the casemates are located the four firemen's wash rooms, coaling-gear machinery rooms, and general mess store and issuing rooms. Abaft and outside the casements to frame 106 are the laundry and the offices of the following: Paymaster, executive officer, engineer, and ordnance, sick bay and medical offices, armory; and on the port side, the post office, chief petty officers' quarters, servants' wash room, ship's store, ordnance storeroom, general mess pantry, printing office and rooms of the chief master-at-arms and the sergeant of marines. Abaft frame 106 are the crew's spaces, wash rooms and water closets.

Berth Deck.—The arrangement of this deck, beginning forward, is as follows: Paint room, chain lockers, junior and warrant officers' storerooms, general store-issuing room, crew space frames 25 to 46. From frames 35 to 46 on the starboard side are located mine-defense outfit, ice-machine room, paymaster's issuing room and clothing and small stores; on the port side are prisons and detention cells. From frames 46 to 77, outboard on either side, are the upper coal bunkers; amidships are the uptake spaces, cold-storage compartment, ice-machine rooms, evaporator room, and machine shop. Abaft frame 95 to frame 122 are crew spaces. From frame 122 to the stern is the lumber and pipe storeroom.

Protective Deck.—This deck extends from frame 19 to frame 112 on a level with and forming the berth deck. Forward of frame 19 it forms the forward part of the upper platform. Aft of frame 112 it slopes down to a point below the water line. The deck is flat elsewhere.

Upper Platform Deck.—This deck extends from frames 19 to 112, except in way of boiler and engine rooms, and on it from forward aft are located stores, windlass engine, magazines, handling rooms, central station, blower rooms and coal bunkers.

Lower Platform Deck.—This deck extends from bow to stern, except in way of boiler and engine rooms. From frames 3 to 11 are the forward trimming tanks. From frames 11 to 25 are sails and awnings storeroom and torpedo room From frames 25 to 46 are magazines and dynamo rooms. From frames 46 to 71½ are the boiler compartments. From frames 71½ to 80 are magazines and storerooms. From frames 80 to 95 are the engine rooms and storerooms. From frames 46 to 71½ on either side, outboard, are the lower coal bunkers. From frames 95 aft to 112 are magazines and storerooms and the steering-engine room. From 112 aft are the sanitary-pump room and the steering-gear rooms.

Hold.—Between frames 3 and 11 are the forward trimming tanks; 112 aft are the after trimming tanks; frames 11 to 19, fresh-water tanks; 19 to 46, dynamo room and storerooms; 46 to 71½, boiler compartments and coal bunkers; 71½ to 80, storerooms; 80 to 95, engine rooms and storerooms; 95 to 112, shaft alleys and storerooms.

Double Bottoms.—The double bottoms extend from frames 11 to 112. In ten of these, under the boiler compartments, is carried reserve feed water; under the engine room are ten fuel-oil double bottoms. All other double bottoms are dry.

BATTERY.

The main battery consists of ten 12-inch guns arranged in pairs, in five turrets on the center line of the vessel. Turrets Nos. 1 and 2 are on the forecastle deck, the latter being ele-

vated so as to fire ahead over the top of the former. Turrets Nos. 3, 4 and 5 are located on the main deck abaft of the smoke pipes. Of this group turret No. 3 is elevated and Nos. 4 and 5 are placed back to back at a lower level well astern of the former.

A secondary battery of sixteen 5-inch rapid-fire guns for torpedo defense is also provided, together with the following smaller guns:

4 3-pdr. saluting guns;
2 1-pdr. semi-automatic guns for boats;
2 0.30-inch machine guns for boats;
2 3-inch field pieces.

There are also two 21-inch submarine torpedo tubes.

BOATS CARRIED.

2 50-foot steam cutters;
1 40-foot steam cutter;
1 40-foot gasoline motor barge for admiral;
2 36-foot sailing launches;
2 33-foot sailing launches;
1 31-foot racing cutter;
2 30-foot cutters;
2 30-foot whale boats;
2 24-foot dinghies;
2 14-foot punts.

ANCHOR WINDLASS.

The anchor engine is of the horizontal type and manufactured by the American Ship Windlass Co. It has two vertical shafts driven by a worm gearing direct from a worm on the crank shaft of the engine. Each shaft has on its upper end, above the forecastle deck, a wildcat, and the arrangement is such that the wildcats can be operated together or independently of each other.

The windlass engine is a double engine, the diameter of the cylinders being 16 inches and the stroke is 16 inches. The working steam pressure is 150 pounds, but the engine is designed to stand full boiler pressure.

STEERING ENGINE AND GEAR.

The steering-engine room is located on the starboard side of the center line, just abaft frame 95, access being from the starboard engine room. The engine is vertical double, with 17 by 14-inch cylinders, constructed by the Williamson Bros. Co., Philadelphia, Pa., and is controlled from the steering stands in the conning tower, on the upper conning platform and in the central station through wire-rope transmission, and also by telemotor.

The engine may also be controlled by a hand wheel in steering-engine room and from the hand-steering room. The various controls are connected by clutches, so that any one may be operated independently of the others. The steering-engine shaft is continued aft to the hand-steering room aft of frame 115, where are located four large wheels for hand steering, provided with suitable clutch, so the handwheels may be disconnected when not in use.

The tiller room is immediately abaft the hand-steering room, the main steering gear consists of a right-and-left-handed screw on which are two driving nuts direct connected by side rods to the crossheads on the rudder stock. The screw is operated through gearing by either the steering engine or the handwheels, either of which may be disconnected by clutches.

COALING ENGINES.

There are two steam-driven coaling engines manufactured by the American Ship Windlass Co., located at Frame No. 62 on gun deck on the port and starboard sides of the ship. These engines are of the vertical type, double cylinder, the diameter of the cylinders being 9-inch and the stroke is 9 inches. The working pressure is 150 pounds, but these engines are designed to stand full boiler pressure, and are fitted with governor gear to regulate the speed.

From each engine through miter gears a 5-inch shaft runs forward (its size there being 3½ inches), and to them are geared five winch heads, which can be thrown in or out at

will by friction clutches. These shafts are cross connected by a shaft and miter gearing so that either coaling engine can actuate all winch heads.

MISCELLANEOUS POWER MACHINERY.

1 motor-driven clothes-washing machine;
1 motor-driven reverse body ironer;
1 motor-driven extractor;
1 motor-driven combined ironer;
1 motor-driven potato peeler;
1 motor-driven dough mixer;
1 motor-driven meat chopper and grinder;
1 motor-driven ice-cream freezer;
1 motor-driven dish-washing machine;
2 motor-driven deck winches, 1 aft and 1 forward;
2 motor-driven fresh-water pumps forward;
2 motor-driven flushing pumps (salt water) aft;
2 motor-driven electric cranes, one on each side of the ship at frame 62, and fitted with suitable motor operating gear;
5 motor-driven turret F. D. blowers;
2 motor-driven torpedo air compressors;
2 motor-driven 5-inch ammunition conveyors;
20 turret-training motors;
10 turret upper-hoist motors;
10 turret lower-hoist motors;
10 turret elevating motors;
10 turret rammers.

DESCRIPTION OF THE FIRE MAIN.

The fire main in the engine and firerooms is a steel, lead-lined pipe, running on each side of the ship, continuously, through each engine room, steam-pipe passage and fireroom, under the protective deck. Six-inch flooding valves and piping are fitted for all lower inboard and outboard bunkers, which may be operated from firerooms or from the ammunition passages, and from compartments B-100 and B-101, on the berth deck.

Forward from the firerooms, the fire main continues through wing passages on each side to frames 26, where there is a cross connection. A 4-inch branch leads forward on the starboard side to the forward pump room. One "fire and bilge" pump discharge pipe connects to each fire main in each fireroom, by means of a 5-inch pipe branching to both sides; these fireroom pumps also connect to the flushing system, by separate valves and pipes in compartment B-100 (ammunition passage), at frames 53, 63 and 71. Six-inch valves and pipes connect the fire main to both distiller pumps in the engine rooms at frame 80, and 6-inch valves and pipes connect the fire main to the engine room "fire and bilge" pumps in both engine rooms.

The following is a list of connections to fire plugs, etc.:

Compartment.	Frame.	Size.	Remarks.
		inches.	
A-100	14	4	Relief valve set at 100 pounds.
A-49	22	1¼	To torpedo air compressors.
A-51	26	6	Connection to sprinkler system.
A-52 S.	25-26	4	Riser to plugs, Nos. 27-44-57-58, and forward relief valve.
A-52 S.	25-26	4	Riser to plugs 15-29-45-59.
A-52 P.	25-26	4	Riser to plugs 14-28-46-60.
A-52 S.	25-26	2½	Hose connections.
A-52 P.	25-26	2½	Hose connections.
A-52 S.	25-26	6	Forward branches join—cut out.
A-52 S.	25-26	6	Forward branches join—cut out.
A-52 P.	25-26	4	Cut out, leading forward.
A-52 S.	26	6	Cross connection—cut out.
A-52 P.	26	6	Cross connection—cut out.
A-66 P. and S.	43	2½	Hose connection to 5-inch ammunition passage.
Dyn. Rm. Trunk	46	4	Cut out to risers.
B-1 P. and S.	50	2½	Cut out to risers.
B-2 P. and S.	61	4	Cut out to risers.
B-3 P. and S.	71	3½	Cut out to risers.
C-1	85-90-93	6	Flooding.
C-2	80-81	5	To sprinkler system.
C-2	85-90-93	6	Flooding.
C-1	88	3½	Cut out.
C-2	88	3½	Cut out.
D-53 S. and P.	110-111	6	After branches joined—cut out. Sprinkler system.
Wiring passage, D-59	108	5	Riser to flushing system.
Wiring passage, D-52	103	2½	Cut out to riser.
Wiring passage, D-60	108	2½	Cut out to riser.
D-54	111	3½	To hose connections in steering-gear compartment D-35.
D-111	107-180	4	Relief valve, lift 100 pounds.

Aft the fire mains run outboard and aft through wiring and wing passages, and join amidships, between frames 110 and 111, with a hose-connection branch leading aft to steering-gear compartment, D-35, frame 115.

A 5-inch valve and pipe connects fire main with flushing system at frame 46, berth deck, port, compartment A-110. Starboard and port engine room "fire and bilge" pumps connect with flushing system at frame 80, compartment B-100. There is a branch to fuel-oil filling pipe on the gun deck at frames 80 and 81.

Fire-main relief valves are fitted between frames Nos. 107 and 108 berth deck, D-111 starboard, and at frame No. 14, in compartment A-100. They are set at 100 pounds, and their discharges lead overboard via scuppers.

SANITARY AND FLUSHING SYSTEM.

The flushing system forward consists of a 5-inch steel, lead lined pipe between frames No. 46 and 81, in compartments B-100 and C-100, forward of frame No. 46, reduced to 3 inches; a branch leads to forward ice machine, wardroom, junior officers' and warrant officers' bath room. Branches lead to cabin bath rooms, to chief petty officers' and firemen's wash rooms, galleys, bakery, ash chutes, dish-washing room, sick-bay baths, laundry and fuel-oil filling line. This system is supplied by four engine-room fire and bilge pumps and the three fireroom fire and bilge pumps. A 5-inch by-pass valve from fire main connects with this system at frame No. 46.

The flushing system aft consists of two electrically-driven centrifugal pumps located in compartment D-34, with a 5-inch discharge pipe leading up to crew's toilet and wash rooms on gun deck between frames No. 117 and 127. A 5-inch by-pass valve from fire main connects with this system at frame No. 108, compartment D-113.

Salt water for admiral's, captain's, chief of staff's and executive officer's baths: valves located at frame 49, compartment B-100 (ammunition passage). Salt water for crew's galley turns on at frame 68, compartment B-100. Salt water for

firemen's wash rooms is taken from the flushing system, between frames 60 and 61, in compartment B-100; its valve being fitted on gun deck. Branches lead to officers' galley, bakery, and portable ash chutes.

FRESH-WATER SYSTEM.

From distiller pumps the fresh water is discharged into fresh-water tanks A-3, 4, 5 and 6 in forward hold, or may be discharged into laundry, chief petty officers' or firemen's wash rooms or overboard. From fresh-water tanks A-3, 4, 5 and 6 the water is pumped to gravity tanks located abaft and higher than the admiral's cabin at frames Nos. 44 and 45, by means of two electric-driven pumps placed in compartment A-46. From gravity tanks water flows to all officers' bath rooms, pantries, galleys, bakery, chief petty officers' and firemen's and servants' wash rooms, general mess pantry, laundry, sick bay, baths, battle dressing stations, and crew's wash rooms. In case the gravity tank is out of order all parts of ship supplied by it may be supplied by one electric pump from compartment A-46. The chief petty officers', firemen's and servants' wash rooms and laundry are fitted with storage tanks from which they take their supply. Other places are supplied directly from the fresh-water main or branches. For filling ship's tanks or reserve-feed tanks from water boats, etc., with fresh water, filling pipes are installed at frames Nos. 25 and 26, and 58 and 59. Each is fitted with by-pass to fresh-water main.

DRAINAGE SYSTEM.

The following is a general description of the drainage system, the features covered being as follows:

1. 15½-inch main drain.
2. 5½-inch secondary drain.
3. 4⅝-inch double-bottom drain.
4. 4½-inch forward bilge drain.
5. 4⅝-inch after bilge drain.
6. 5-inch independent drain fireroom (Bu. of S. E.).
7. 5-inch independent drain engine rooms (Bu. of S. E.).

Main Drain.—It is 15½ inches in diameter throughout its entire length and is located on the starboard side outboard. It extends from the after end of the forward fireroom to the forward end of the engine room, where it branches and runs athwarthships, each branch connecting to a centrifugal pump.

A 15½-inch stop-check valve is located in each fire and engine room, forming part of the main system, and all are operated either at the valves or on the berth deck through deck plates.

In the port and starboard engine rooms at frame 84, there are 5-inch connections between the main drain and C. & R. manifolds, located at frames 83–84. These manifolds are connected to the fire and bilge pumps in the fire and engine rooms.

Secondary Drain.—It is 5½ inches in diameter throughout its entire length, frames 48–92, and has connections to all fire and bilge pumps through C. & R. and S. E. manifolds; also to handy-billy-pump manifolds, located on upper platform deck, forward at frame 46 and aft at frame 95. The forward connection is made directly to the secondary drain and the after connection is made through the after bilge suction located in the starboard engine room at frame 93.

The forward end of the secondary drain connects to the 4½-inch forward-bilge drain, located in forward fireroom, frame 50–51, and at the after end to the after bilge suction at frame 92.

There is a 5½-inch connection from the bilge wells in each fire and engine room to the secondary drain.

Double-Bottom Drain.—It is on the port side of the firerooms, between frames 46 and 71, and is divided into three sections, each being 4⅝ inches in diameter where suctions to independent double bottoms are 3 inches in diameter. It drains twelve double bottoms and has a connection to one C. & R. manifold in each fireroom.

The double-bottom compartments are flooded through S. E. manifolds connected to C. & R. manifolds by operating their respective stop-check lift valves.

Oil Double Bottoms.—The double bottoms for oil-fuel stowage are under the engine rooms and have independent suctions to the two oil-fuel pumps, one in each engine room.

Forward-Bilge Drain.—It is 4½ inches in diameter and drains all the double bottoms forward of frame 41 and compartment A–11. The latter is the drainage tank for chain lockers, handling rooms, forward torpedo room and Nos. 1 and 2 turrets. This drain ends at frame 19 in a manifold, from which suctions are led to the forward compartments and the trimming tanks.

After-Bilge Drain.—The 4⅝-inch after-bilge drain connects with the secondary drain, at frame 91, in starboard engine room, draining the " D" compartments and the after trimming tanks, where it is 3 inches in diameter.

Independent Drains.—These drains come under the cognizance of Bureau of Steam Engineering. In each fireroom they are 4⅛ inches in diameter and run from the bilge wells to the fire and bilge pumps.

The independent drains in each engine room are 5 inches in diameter. They drain the bilge wells in shaft alleys and engine rooms. The drains in the shaft alleys connect to S. E. manifolds, in port and starboard shaft alleys, frames 96, and from there have a direct connection to fire and bilge pumps in the engine rooms.

C. & R. Manifolds.—Drainage System.

Location, frame.	Port or starboard.	Compartment.
19	Port.	A–8
52	C.L.	B–1
57	C.L.	B–2
68	C.L.	B–3
84	Starboard.	C–1
84	Port.	C–2
46	Starboard.	A–64 S.
95	Starboard.	D–38

TRIMMING TANKS.

There are four trimming tanks, two forward and two aft. The forward tanks are flooded by a 6-inch sea connection at frame 11, starboard side. A 4-inch pipe is led to each trimming tank, where a 4-inch globe valve is operated at the valve and on the berth deck in deck plates, frame 11.

The after tanks are flooded by a 6-inch sea connection, frame 112, starboard side; a 4-inch pipe is led to each trimming tank, where a 4-inch globe valve is operated at the valve and on the berth deck, in deck plates, frame 112.

Capacities of Double-Bottom Compartments.

Compt.	Frames.	Side.	Cubic feet.	Tons F.W.	Tons S.W.
A-92	11-15	Across	1,594.8	44.5	45.83
A-93	15-19	Across	1,856.1	51.8	53.35
A-94	19-25	Across	3,373.	94.1	96.91
A-95	25-31	Between wings	2,949.4	82.3	84.73
A-96	31-35	Between wings	2,136.1	59.6	61.38
A-97	35-41	Between wings	3,383.5	94.4	97.21
A-98	41-46	Center to port wing	1,881.4	52.5	54.07
A-99	41-46	Center to stbd wing	1,881.4	52.5	54.07
B-80	46-50	Port	687.8	19.2	19.78
B-81	46-50	Starboard	687.8	19.2	19.78
B-82	50-55	Port	924.6	25.8	26.57
B-83	50-55	Starboard	924.6	25.8	26.57
B-84	55-59	Port	716.8	20	20.6
B-85	55-59	Starboard	716.8	20	20.6
B-86	59-63	Port	716.8	20	20.6
B-87	59-63	Starboard	716.8	20	20.6
B-68	46-50	Port wing	1,182.5	26.2	27
B-69	46-50	Starboard wing	1,182.5	26.2	27
B-70	50-55	Port wing	1,571	35.9	37
B-71	50-55	Starboard wing	1,571	35.9	37
B-72	55-59	Port wing	1,351.6	31.1	32
B-73	55-59	Starboard wing	1,351.6	31.1	32
(Wing double bottoms B-68 to B-73, inclusive, can be filled only to upper manhole—completely filled they hold 51 tons more of salt water.)					
D-97	95-100	Port and starboard	2,326	64.9	66.82
D-98	100-105	Port and starboard	2,161	60.3	62.11
D-99	105-110	Port and starboard	3,207.5	89.5	92.18

Capacities of Trimming Tanks.

A-1	Stern-6	Side forepeak	3,512.5	98.06	101
A-2	6-11	Side forepeak	5,533	154.4	159
D-8	112-118	Forward after TT	3,369.7	94.18	97
D-9	118-stern	After TT	1,111.1	31.07	32

(Note.—Fresh water, 34.8 cu. ft. to ton; salt water, 33.79 cu. ft. to ton.

Capacities of Reserve Feed-Water Tanks.

Compt.	Frames.	Side.	Cubic feet.	Tons F.W.	Tons S.W.
B-88	46-50	Port	874.5	24.4	25.13
B-89	46-50	Starboard	874.5	24.4	25.13
B-90	50-55	Port	1,093.1	30.5	31.42
B-91	50-55	Starboard	1,093.1	30.5	31.42
B-92	55-59	Starboard	874.5	24.4	25.13
B-93	55-59	Port	874.5	24.4	25.13
B-94	59-63	Starboard	874.5	24.4	25.13
B-95	59-63	Port	874.5	24.4	25.13
B-96	63-67	Starboard	1,663	46.4	47.79
B-97	63-67	Port	1,663	46.4	47.79

Capacity of Fresh-Water Tanks.

Tank No.	Compartment.	Between frames.	Location.	Side.	Capacity.	
					Gallons.	Tons.
A-3	A	11-15	Hold	Starboard	6,103	22.7
A-4	A	11-15	Hold	Port	6,103	22.7
A-5	A	15-19	Hold	Starboard	6,103	22.7
A-6	A	15-19	Hold	Port	6,103	22.7
Gravity Tanks.						
No. 1	...	...	Main deck.	Starboard.	1,000	3.7
No. 2	...	...	Main deck.	Port	1,000	3.7
Total,					26,412	98.2

DECK DRAINS.

For sanitary purposes all drains from baths, toilets, wash rooms, pantries and galleys drain overboard directly through pipes by the shortest route, and are located as follows:

Name and use.	Frame.	Side.	Drain.	Scupper.
			inches.	*inches.*
Ward room and warrant officers' baths and toilets.	11-12	Port	5	9 × 3½
Junior officers' baths and toilets	11-12	Starboaad	5	9 × 3½
Ward room, junior officers' and warrant officers' pantries	22-23	Port	2	9 × 3½
Admiral's and captain's baths and pantries	47-48	Port	4	9 × 3½
Firemen's wash rooms	56-57	Std. & port.	2½	9 × 3½
Officers' galley and bakery	56-57	Std. & port.	2	9 × 3½
Crew's galley	64-65	Std. & port.	2	9 × 3½
Firemen's wash room	64-65	Std. & port.	2½	9 × 3½
Chief petty officers and servants' wash room	94	Port	2	9 × 3½
General mess pantry	90	Port	2	9 × 3½
Laundry	75-76	Starboard	2	9 × 3½
Sick bay	103	Starboard	4	9 × 3½
Crew's wash rooms and toilets	120-121	Port & std.	5	9 × 3½
Crew's wash rooms and toilets	123	Port & std.	5½	9 × 3½

MAGAZINE FLOODING AND SPRINKLING.

There are three flooding systems for the three groups of magazines, forward, middle and after. Each system floods the magazines of its group on the lower platform deck, while the upper platform magazines are flooded or sprinkled by water from the fire main.

The forward and after systems are each composed of two separate systems, port and starboard, each of which is supplied by 8-inch sea connections, frame 31, forward, and frame 105, aft.

The middle system has its 9-inch sea connection between frames 72–73 and floods the middle group of magazines.

These flooding systems run above the lower platform beams forward, and below them for the middle and after groups, and branch pipes, are led up through the decks to valves in magazine spaces.

Adjacent to the sea connections are globe valves operated at the valves on the berth deck in deck plates.

Upper-Platform Systems.—These are connected to the fire main as described thereunder, and in other particulars have the characteristics of the normal flooding system. All magazine-flood valves are gate valves operated on the berth deck in deck plates.

Sprinkling Systems.—A sprinkling system is fitted in all magazines and shell rooms and consists in the magazines of 2½-inch brass pipe perforated on the under side, so that each powder tank can be sprinkled, and is fitted with cut-out valves in each magazine. The latter are operated at valve and on the berth deck in deck plates. This system is connected with the fire system which serves the flooding system on the upper platform.

MAGAZINE COOLING.

For all the powder magazines except those containing saluting powder and small-arms ammunition, provision is made for supplying artificially-cooled air. There are three systems supplying, respectively, the forward, the middle and the after

group of magazines, each system being supplied by its own ventilating blower; the two forward blowers having a capacity of 2,500 cubic feet per minute and the middle 1,600 feet.

The method of cooling the air is as follows: Compressed air is led from the Allen dense-air ice machines to a Schutte-Koerting corrugated-film air cooler, where, on expanding, heat is extracted from the incoming air on its way to the magazines from the ventilating blowers.

Forward and aft the coolers are of three units and amidships of two units.

The magazines are lagged where necessary to prevent the conduction of heat.

The cooling capacity per unit is 800 cubic feet of air per minute from 70 degrees to 50 degrees F., when the dense air enters the cooler at 50 degrees F. and leaves at 30 degrees F. with sea water at 70 degrees F.

REFRIGERATING PLANT.

There are four Allen dense-air ice machines, each capable of producing the cooling effect of three tons of ice per day.

One ice machine is located forward, primarily for cooling the forward magazines, but can be used on the cold-storage system if desired. The other ice machines are located just aft of the engine-room hatches, one on the starboard and two on the port sides, and are fitted for cold-storage service, ice making, etc., and for the midship and after magazines cooling systems.

There are five refrigerating rooms, isolated by air locks and insulated with cork in the usual manner.

VENTILATING SYSTEM.

Artificial ventilation has been provided where necessary for all quarters, living spaces, passages, storerooms and magazines. There are thirty-two ventilating motors and fans, each on its own circuit. In general the supply of air is on the plenum system to the different compartments, the air being

renewed at a previously determined rate according to the requirements. The toilet spaces are ventilated on the exhaust system. The ventilating fans are manufactured by the American Blower Co., Detroit, Mich., and the motors by the Diehl Mfg. Co., Elizabethport, N. J. They are located as shown in the following table.

No.	Compartment.		Deck.	Side.	Frame.	Remarks.
	No.	Name.				
1	A-118	Lamp room	Main.	P.	4-5	.....
2	...	Turret No. 1	...	...	...	On upper handling-room level.
3	...	Turret No. 2	...	...	...	Do.
4	B-116	.....	Gun...	P.	44	.....
5	B-116	Crew space	Gun...	S.	44	.....
6	B-116	Crew space	Gun...	S.	47	.....
7	B-116	Crew space	Gun...	P.	58-60	.....
8	B-116	Crew space	Gun...	P.	69-70	.....
9	C-113	Crew space	Gun...	S.	74	.....
10	C-113	Crew space	Gun...	P.	78	.....
11	C-113	Crew space	Gun...	S.	78	.....
12	...	Turret No. 3	...	...	...	Do.
13	C-113	Crew space	Gun...	S.	79	Magazine cooler.
14	C-113	Skylight trunk	Gun...	A.	91	.....
15	...	Turret No. 4	...	...	...	Upper H. R. level.
16	D-109	Crew space	Berth.	P.	104	.....
17	...	Turret No. 5	...	...	...	Do.
18	A-104	General stores	Berth.	P.	20-21	.....
19	A-108	Crew space	Berth.	S.	35	Magazine cooler.
20	A-108	Crew space	Berth.	P.	32	.....
21	A-110	Crew space	Berth.	S.	35-36	.....
22	C-100	Crew space	Berth.	P.	83	.....
23	C-100	Crew space	Berth.	S.	83	.....
24	D-109	Crew space	Berth.	P.	104	.....
25	D-109	Crew space	Berth.	S.	102	Do.
26	D-109	Crew space	Berth.	S.	101	.....
27	D-113	Crew space	Gun...	S.	124	.....
28	D-112	Lumber and stores	Berth.	S.	114	.....
29	C-43	Blower room	U.P.	S.	83	.....
30	C-44	Blower room	U.P.	P.	83	.....
31	C-43	Blower room	U.P.	S.	84	.....
32	C-44	Blower room	U.P.	P.	84	.....

The air as supplied by these fans, running at their maximum speed, is designed for a pressure of from 6 to 11 pounds to the square foot and velocity of from 1,200 to 2,000 feet per minute, depending on the designed capacity and the purpose for which the compartment is used.

None of the principal longitudinal or transverse watertight

bulkheads of the ship have been pierced by the main ducts. Where branches pass through the protective deck they are made watertight to a point just below the gun deck. All branches passing through magazines are galvanized-steel seamless tubing, or built of heavy-gage steel worked watertight. A natural-exhaust duct, exceeding the area of the mechanical-supply duct, has been provided for each magazine not refrigerated, and located as far as practicable from the supply duct. The upper ends of these ducts are carried up close to the gun deck and terminate there with a gooseneck, the lower end of which is bell mouthed and covered with wire mesh.

McCrary adjustable elbows, fitted with butterfly dampers, have been used for supply terminals in all quarters, living spaces and elsewhere. In the quarters they are nickel plated and elsewhere galvanized. All openings of these elbows are fitted with portable wire mesh.

There are adjustable mushroom cowls which fit in the deck and serve as inlets for the various blowers. Two forward and one aft are the only ones fitted to unship at " general quarters."

The ventilation of the bunkers is exhaust only, and that not forced. Galvanized-steel tubing leads from all bunkers into the uptake air spaces and are fitted with inswinging airtight doors of the damper type; all openings into the bunkers are fitted with wire mesh.

For ventilating the engine rooms there are two blowers, port and starboard. Branches are led from the mains to all working stations, platforms, air casings and corners of the rooms where required for ventilation.

Sheathing.—To retard transmission of heat to adjacent magazines and ammunition passages the pipe passages, fire-rooms, steering-engine room, engine rooms and coal-bunker bulkheads separating the former from the five latter are lagged or sheathed.

From the forward lower coal-bunker bulkheads exhaust piping leads to the air casing under the protective deck in the forward fireroom, whose air in turn exhausts to sheathing in

the fireroom hatch, and from there to the atmosphere. The heated air from the casing under the protective deck of the other two firerooms exhausts similarly, as does the sheathed after athwartship bulkhead of the after fireroom.

The under sides of the protective deck, within the fire and the engine rooms, have been fitted with sheathing, with an air space. This space within the engine rooms is connected with the ventilating system for supply, and to the engine hatches for exhaust. In the firerooms this space is similarly ventilated, the exhaust escaping via uptake air space near the top.

Heated Air.—A new departure in connection with the ventilation of the living quarters is the installation of steam-heating coils in the air-supply pipes. These heating coils in cold weather will raise the temperature of the incoming new air.

HEATING SYSTEM.

The heating system is divided into two sections, one forward and one aft. The forward section is divided into six circuits, one of which is for galley, bakery and firemen's washrooms; two are for bath and pantry circuits, one for radiator circuit in admiral's and captain's quarters, and two for heater or thermo tanks located on berth and gun decks.

The after section is divided into six circuits consisting of one radiator, one washroom and sick-bay bath, and four thermo-tank circuits.

All steam lines to heating systems are fitted with pressure gages, stop and relief valves, set at 30 pounds; from stop valves the steam is carried to distributing manifolds and divided into circuits, each circuit being arranged with a stop valve so that it can be operated independently of the others.

There being no auxiliary steam line in firerooms, all the forward circuits receive their steam from the main steam line direct. The gravity-tank heater coils receive steam from the forward circuit. Steam for the laundry is taken from the after circuits.

The drains from radiators and thermo tanks are also divided into circuits and lead to receiving manifolds, located forward and aft, each circuit being arranged with check and stop valves

to prevent water from one circuit backing up into another. The discharge from these manifolds leads to a steam trap in the port engine room, which discharges into the filter tank, or to main and auxiliary condensers.

The drains from galleys and pantries also lead to traps in port engine room, discharging into filter tanks and the condensers. The drains from bath rooms and other water heaters have no connection with radiator drains but go direct to auxiliary exhaust lines in Nos. 1 and 2 firerooms and in both engine rooms.

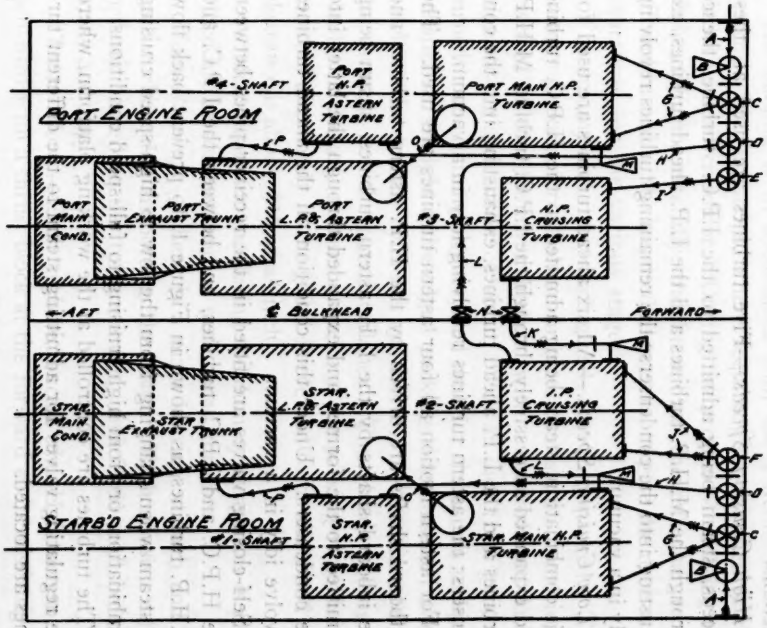
The heater-circuit steam and drain pipes consist of seamless-drawn brass pipe, iron-pipe size; all pipes up to 1½ inches being connected together by composition fittings, but above that size composition flanges are used. Connections at watertight bulkheads are made with composition stuffing boxes, and copper U bends are installed throughout to provide for expansion.

The radiators consist of coils made up of 1-inch seamless-drawn brass pipe, iron-pipe size, composition fittings, stop and air valves.

The number of heater or thermo tanks fitted throughout the berth and gun decks are 21, and consist of casting or header with 1-inch seamless-drawn brass-pipe coils, varying in number from 7 to 22, according to size of box. These boxes are of the vertical and horizontal types, and fit between deck beams, or in the case of the larger size stand on a frame on the gun deck. These boxes are fitted to receive air from ventilating fans and pass it through the steam-filled coils before it discharges into living spaces. The coils are insulated with block magnesia, covered with No. 16 sheet iron. By arrangement of dampers the air may be by-passed in warm weather. In case of a leak any coil in box can be removed without disturbing the others.

MAIN ENGINES.

The propelling machinery consists of Parsons turbines, designed to run at 330 revolutions per minute, when developing 28,000 shaft horsepower. They are arranged on four lines of shafting, as shown in sketch, Figure 1.



MARK	DESCRIPTION OF PART.
A	14" MAIN STEAM FROM BOILERS
B	14" ENGINE STOP AND GOVERNOR VALVE
C	14" REGULATING VALVE, M.H.P. TURBINE
D	13" " " , H.P. ASTERN "
E	10½" " " , H.P.C. "
F	14" " " , I.P.C. "
G	10½" STEAM TO M.H.P. TURBINE
H	13" " " H.P. ASTERN "
I	10½" " " H.P.C. "
J	10½" " " I.P.C. "
K	17" STEAM FROM H.P.C. TO I.P.C.
L	17" " " I.P.C. TO M.H.P.
M	17" SELF-CLOSING VALVE
N	17" CUT-OUT VALVE
O	43" STEAM FROM M.H.P. TO L.P. AHEAD
P	29" " " H.P. AST. TO L.P. AST.

FIG. 1.

SKETCH SHOWING ARRANGEMENT OF MAIN ENGINES AND PIPING.

The arrangement provides six ahead and four astern turbines, each L.P. turbine embodying an astern turbine in the after end of its casing.

For ahead motion the outboard shafts are driven by the M.H.P. turbines, and the inboard shafts by the L.P. ahead turbines alone, or in combination with the I.P.C. and H.P.C. turbines, as described in the following paragraphs.

Full Speed Ahead.—Only four turbines are used for this purpose, steam being admitted to the M.H.P. turbines and expanded through the L.P. ahead turbines into the condensers. Under this condition the astern and cruising turbines revolve idly in a vacuum, which is maintained through the drain connections.

High Cruising Speeds.—Five turbines are used at these speeds, steam being admitted to the I.P.C. turbine, thence through the M.H.P. turbines and the L.P. ahead turbines, exhausting into the condensers; the remaining turbines revolving idly in a vacuum.

Low Cruising Speeds.—All six ahead turbines are used for this combination, steam being admitted to the H.P.C. turbine and expanded successively through the I.P.C. turbine, M.H.P. turbines and the L.P. ahead turbines, exhausting into the condensers; the astern turbines revolving idly in a vacuum.

For astern motion all four astern turbines are used. The outboard shafts are driven by the H.P. astern turbines and the inboard shafts by the L.P. astern turbines, steam being admitted to the former and expanded through the latter into the condensers. Under this condition all the ahead turbines revolve idly in a vacuum.

Self-closing valves are fitted in the receiver pipes between the H.P.C. and I.P.C. turbines, and between the I.P.C. and M.H.P. turbines, as shown in Figure 1, to prevent back flow of steam when changing from the low to high-speed cruising combination, or from high cruising to full-speed conditions.

The turbines are controlled at the working platform, where the regulating valves for admitting steam to the different turbines are located.

Turbines.—Each turbine is composed of two essential parts, first, the fixed part, or cylinder, and second, the moving part, or rotor. In the cylinder are mounted the guide blades and on the rotor the moving blades. The cylinders are of hard, close-grained cast iron, divided into two parts at the axis on a horizontal plane, the lower half being provided with feet for bolting to the seating. Each rotor is built up of a drum of forged steel, securely fastened to a cast-steel wheel at each end, that is shrunk on and keyed to the rotor shaft. Great care was exercised to assure the rotors being perfectly balanced when entirely completed, in order to prevent vibration when running.

The blading is in accordance with the Parsons standard system. Binding strips, where required, are fastened to the blades with silver solder. During the process of caulking the blades, they are carefully straightened and the ends chamfered off and turned to the required diameters, to give the necessary clearances for the various stages.

The dummy and rotor-shaft glands are steam packed. The dummy packing for all ahead turbines consists of brass rings, inserted in the dummy cylinders, with a hook-shaped knife edge having a clearance of about .02 of an inch from grooves turned in the dummy sleeves; and that for the astern turbines consists of straight knife-edged packing rings inserted alternately in the dummy cylinders and sleeves, the clearance being the same as for the ahead turbines. The gland packing is similar to the astern dummy packing, the packing rings being inserted alternately in the gland sleeve and rotor shaft and the clearance is about .015 of an inch.

A micrometer for measuring the dummy clearances is provided at the forward end of each ahead turbine.

Main Bearings.—There is a main bearing at each end of each turbine for supporting the rotor. Each consists of a pedestal bolted to the turbine casing and fitted with bottom brass and cap.

Thrust Block.—Each turbine, except the H.P. astern, is provided with a thrust block at the forward end, consisting of

a number of brass rings, in halves, fitted into corresponding collars on the shaft. The lower half of each bearing is for taking the ahead and the upper half the astern thrust.

All main bearings, thrust bearings and line-shaft bearings are fitted with a closed system of forced lubrication, as described later.

Governor.—A Proell governor is fitted to each line of shafting, the governor mechanism operating the main steam stop valve in the engine room.

Turning Gear.—Each line of shafting has an electrically-operated turning gear, consisting of a 5-horsepower, reversible Diehl motor, which engages through gears and worm the worm wheel on the main shaft. Provision is also made for turning by hand with a ratchet.

Lifting Gear.—An efficient lifting gear, consisting of two 20-ton Sellers Co. trolleys on a track over each shaft in wake of the turbines, is provided. The lifting mechanism is hand-operated through chains from the engine-room floor.

Main Turbine Data.

Rotor drums:	Diameter	Length.
Main H.P., inches	71	114
H.P. cruising, inches	71	63½
I.P. cruising, inches	70	72½
L.P. ahead, inches	97	87⅝
H.P. astern, inches	71	36¼
L.P. astern, inches	71	44⅛
Number of expansions:		
Main H.P. and L.P. ahead, each		6
H.P.C. and I.P.C., each		3
H.P. astern and L.P. astern, each		4
Turbine casings, diameter, inches, each expansion:		
Main H.P.	73½, 74½, 75½, 77½, 80, 83½	
L.P. ahead	107½, 111½, 117½, 128, 128, 128	
H.P.C.	72½, 72⅝, 73⅛	
I.P.C.	72½, 73½, 74½	
H.P. astern	72½, 73½, 75, 77	
L.P. astern	80, 83½, 83½, 83½	
Length of casing for each expansion and diameter noted above:		
Main H.P., inches	17½, 17, 17 1⅜, 18⅝, 19½, 24 9/16	
L.P. ahead, inches	12 1⅜, 13, 15 7/16, 19, 19, 19	
H.P.C., inches	18½, 18½, 25⅛	
I.P.C., inches	23½, 23½, 24⅛	
H.P. astern, inches	9½, 8¼, 8¼, 9	
L.P. astern, inches	9½, 11 15/32, 11 23/32, 11 31/32	

Rows of blading for each expansion :	
Main H.P.	13
L.P. ahead	3 of 7, 3 of 6
H.P.C.	20
I.P.C.	18
H.P. astern	6
L.P. astern	5
Length of blades for each expansion, inches :	
Main H.P.	1½, 1⅞, 2½, 3½, 4½, 6½
L.P. ahead	5½, 7½, 10½, 13½, 13½, 13½
H.P.C	⅝, 1⅜, 1 1/16
I.P.C.	1½, 1⅞, 2⅛
H.P. astern	⅞, 1⅜, 2, 3
L.P. astern	4½, 6½, 6½, 6½

Rotor shaft and bearings :	Length, inches.	Diameter, inches.	Diameter axial, inches.	Length over all of rotor drum and shaft, feet and inches.
Main H.P.	18½	14	9	21–07½
L.P. ahead	29	15	11	25–10¾
H.P.C.	14½	14	11	15–02½
I.P.C.	14½	14	11	16
H.P. astern	12¾	14	9	11–02½
L.P. astern	29	15	11	With ahead.

Thrust bearings :	M.H.P., each.	L.P. ahead, each.	H.P.C., each.	I.P.C., each.
Collars on shaft, number	17	17	8	8
Thickness, inch	0¾	0¾	0¾	0¾
Distance between, inches	1 1/16	1 1/16	1 1/16	1 1/16
Outside diameter, inches	17½	17½	17½	17½
Inside diameter, inches	12½	12½	12½	12½
Number of shoes, top	16	16	7	7
bottom	17	17	8	8

SHAFTING.

There are four lines of shafting, a pair port and starboard respectively. Each pair is parallel in itself, but diverges from the center line of the vessel and slopes downward aft.

The outboard shafts are in two sections each, consisting of one line shaft supported by a spring bearing and a propeller shaft, extending through the stern tube and supported by the strut and stern-tube bearings. The inboard shafting is in four sections each, there being two line shafts supported by three spring bearings, one stern-tube shaft carried by the stern-tube bearings, and a propeller shaft supported by one strut bearing.

All stern-tube and strut bearings are lined with lignum vitæ and the shafts are composition bushed at these bearings. The

shafting within the stern tube is covered with a composition casing.

The inboard coupling consists of a sleeve secured by four keys to the stern-tube shaft. Back of the sleeve is a collar made in halves and secured to the sleeve and to the coupling disc on the line shaft by fitted bolts.

The outboard coupling is of the split-sleeve type, consisting of two half sleeves secured to each shaft by two keys; the half sleeves being secured together by bolts.

Shaft Data.

Line shaft, diameter outside, inches	12⅛
axial hole, inches	6
Stern-tube shaft, diameter outside, inches	12⅜
axial hole, inches	6
Propeller shaft, diameter outside, inches	12⅜
axial hole, inches	6
Couplings, diameter, inches	22
thickness, inches	3
Inboard coupling, diameter of sleeve outside, inches	22½
inside, inches	14⅛
length of sleeve, inches	12
thickness of collars, inches	2⅜
Coupling bolts, number each coupling	8
diameter (taper)* at face of coupling, inches	2⅜
Outboard coupling, length of half sleeves, inches	48
bolts securing sleeves, number	16
diameter, inches	1¾
Spring bearings, diameter, inches	12⅛
length, inches	18
Forward stern-tube bearings, diameter, inches	14⅝
length, inches	49½
After stern-tube bearings, diameter, inches	14⅝
length, inches	60
Strut bearings, diameter, inches	14⅜
length, inches	72

PROPELLERS.

There are four three-bladed propellers, all outboard turning when going ahead. The blades and hubs are of monel metal and cast in one piece. The blades are true screw machined to pitch.

*Parallel bolts for inboard coupling.

Propeller Data.

Diameter of propeller, inches	110
hub, inches	27
Pitch, inches	102
Ratio of pitch to diameter	.927
Area, projected, square inches	5,307
helicoidal, square inches	5,841
disc, square inches	9,503
Ratio, projected to disc area	0.56
helicoidal to disc area	0.61

MAIN CONDENSING APPARATUS.

Main Condensers.—There is one main condenser in the after end of each engine room, of the following principal dimensions:

Inside diameter, feet and inches	10–00
Thickness of shell (steel), inch	00½
Length between tube sheets, feet and inches	11–00
Thickness of tube sheets, inch	01
Number of tubes	8,466
Diameter of tubes, inch	00⅝
Thickness of tubes, B. W. G., No.	16
Cooling surface, square feet	15,235
Exhaust nozzle, area, square feet	48
Diameter of air-pump suction, inches	12
augmenter suction, inches	15
circulating water inlet and outlet, inches	30

Vacuum Augmenter.—In order to obtain the maximum possible vacuum, so essential to efficient operation of turbine machinery, a Parsons vacuum augmenter is installed for each main condenser. The apparatus consists of a small condenser of 450 square feet cooling surface placed below and connected to the bottom of the main condenser by a pipe having a conical contracted portion, through which a jet of steam is forced. The end of this pipe extends inside the main condenser shell to a height of about 6 inches above the bottom, which prevents the water of condensation from entering it. The steam jet exhausts most of the air and vapor from the condenser and delivers it to the air-pump suction via the augmenter condenser. A water seal is placed in the air-pump

suction, between the suction from the augmenter condenser and that from the main condenser, which prevents the air and vapor thus removed from returning to the main condenser. Figure 2 shows a sketch of this apparatus.

**Main Air Pumps.*—An air pump, of the M. T. Davidson, vertical, twin, bucket, single-acting type, is provided for each main condenser. The steam cylinders are 17½ inches diameter and the water cylinders 35 inches diameter with a common stroke of 21 inches. The suction nozzle is 12 inches and the discharge nozzle 10 inches.

**Main Circulating Pumps and Engines.*—There is one centrifugal circulating pump for each main condenser, driven by a compound engine. The principal dimensions of pump and engine are as follows:

Capacity of pump, gallons per minute	15,000
Diameter of suction nozzle (two), inches	21
discharge nozzle, inches	30
impeller, inches	51
H.P. cylinder, inches	11½
L.P. cylinder, inches	23
Stroke, inches	12

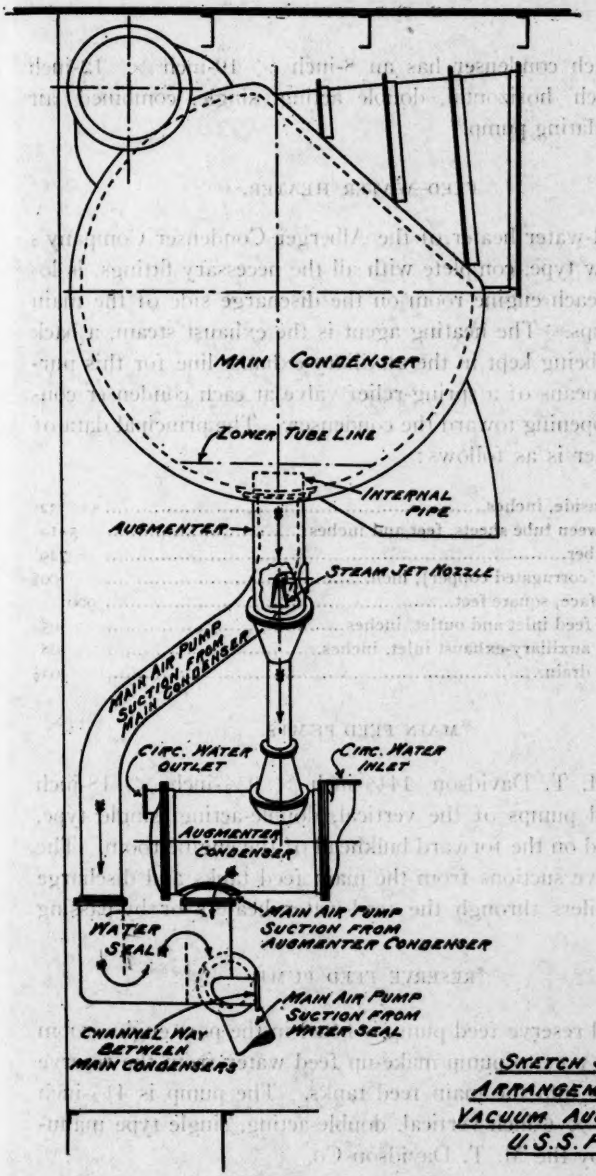
FEED AND FILTER TANK.

A feed and filter tank of 4,000 gallons capacity is located at a high level in the forward inboard corner of each engine room. The filter chamber is in the top of the tank and has a capacity of about 700 gallons. The filter has an inner bottom of loose perforated plates and is divided into compartments, in which is placed the filtering material, by vertical division plates. These partitions are so arranged that the water in passing through the filter will flow under and over in succession.

AUXILIARY CONDENSER.

In each engine room there is an M. T. Davidson auxiliary condenser of 750 square feet cooling surface, connected through the auxiliary exhaust pipe to all the auxiliary machin-

* See also Table I.



SKETCH SHOWING ARRANGEMENT OF VACUUM AUGMENTER; U.S.S. FLORIDA.

FIG. 2.

ery. Each condenser has an 8-inch × 10-inch × 12-inch × 12-inch horizontal, double acting, single, combined air and circulating pump.*

FEED-WATER HEATER.

A feed-water heater of the Alberger Condenser Company's triple-flow type, complete with all the necessary fittings, is located in each engine room on the discharge side of the main feed pumps. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary-exhaust line for this purpose by means of a spring-relief valve at each condenser connection, opening toward the condenser. The principal data of each heater is as follows:

Diameter, inside, inches	32
Length between tube sheets, feet and inches	5–10
Tubes, number	749
size (corrugated copper), inch	00¾
Heating surface, square feet	900
Diameter of feed inlet and outlet, inches	05
auxiliary-exhaust inlet, inches	08
drain	03½

*MAIN FEED PUMPS.

Two M. T. Davidson 14½-inch × 9½-inch × 18-inch main feed pumps of the vertical, double-acting, single type, are located on the forward bulkhead of the engine room. The pumps have suctions from the main feed tanks and discharge to the boilers through the feed-water heaters or by-passing same.

*RESERVE FEED PUMP.

A small reserve feed pump is fitted in the port engine room for use in port to pump make-up feed water from the reserve feed tanks into the main feed tanks. The pump is 4½-inch × 5-inch × 6-inch vertical, double-acting, single type manufactured by the M. T. Davidson Co.

* See also Table I.

*ENGINE-ROOM FIRE AND BILGE PUMPS.

Two M. T. Davidson 12-inch × 10-inch × 18-inch vertical, double-acting, single, fire and bilge pumps are provided in each engine room. They are arranged to draw water from the bilge, drainage system and sea, and discharge to the fire main, sanitary system and overboard.

*PIPE INSULATOR PUMPS.

In each engine room is a 6-inch × 8-inch × 8-inch vertical, double-acting, single pump of the M. T. Davidson pattern, for circulating water around the main steam pipe flanges at bulkheads near magazines to prevent the transmission of heat through the ship's structure to the magazines.

WATER SERVICE.

The water service consists of 1½-inch connections from the fire and bilge-pump discharge to the oil-cooler circulating water and from the fire main.

No permanent connections are fitted; conveniently located hose connections alone being provided in the engine rooms and shaft alleys for the bearings and parts that may require cooling.

There are also connections for flushing the stern-tube bearings.

FORCED LUBRICATION.

An elaborate system of forced lubrication is provided in each engine room for the main, thrust and line-shaft bearings and the main circulating pump engines. It consists of two oil pumps,* two oil coolers, one circulating pump for the oil coolers, one 340-gallon oil-drain tank and the necessary piping and fittings.

The operation is as follows: The oil pumps draw oil from the drain tank and deliver it via the oil coolers to the various

* See also Table I.

parts to be lubricated at a pressure of about 15 pounds. Macomb strainers are fitted in both suction and discharge connections to and from the pumps, and a by-pass is arranged around the oil coolers for emergency use. After the oil has passed through the bearings it is caught in troughs formed in the bearing bases and drained by gravity back to the drain tank, from whence the cycle is repeated.

Lock cocks are fitted at each oil inlet to bearings, for regulating the supply, and sight glasses for observing the oil flow are fitted in the drain from each bearing. A thermometer is also fitted at each sight glass to show the temperature of the oil leaving the bearings.

In addition to the oil-cooler circulating pump provision is made for circulating cooling water through the oil coolers by one of the fire and bilge pumps in the engine room.

Oil storage tanks of 1,000 gallons capacity are provided in each engine room for replenishing the system, etc.

BOILERS.

The boilers, twelve in number, are of the Babcock & Wilcox water-tube type, arranged in batteries of four each in three separate watertight compartments. They are designed to run the entire machinery installation at full power, with an average air pressure in the ash pits of not more than two inches of water.

The uptakes are of the usual design, and there are two smoke pipes, each about 92 feet in height above the grates and 11 feet 6 inches in diameter.

The following internal fittings are provided for each boiler:

1 6-inch dry pipe;
1 2½-inch main feed pipe;
1 2½-inch auxiliary feed pipe;
1 1½-inch pipe and scum pan for surface blow;
Zinc protectors and buckets.

The following external fittings are provided for each boiler:

1 5½-inch steam stop valve;
1 triple safety valve—each valve 4 inches in diameter;
2 2½-inch stop valve for main and for auxiliary feed;
2 2½-inch check valve for main and for auxiliary feed;
2 water gages;
1 1½-inch surface-blow valve;
2 1½-inch bottom-blow valves;
1 steam gage;
3 gage cocks;
1 air cock;
1 salinometer pot connection;
1 connection for testing water in boilers.

Boiler Data.

Number	12
Working pressure, pounds per square inch	200
Test pressure, pounds per square inch	300
Height to top of drum, feet and inches	13–10¾
Length on floor, feet and inches	9–01½
Width on floor, feet and inches	18–04½
Drum, diameter, inches	42
length, feet and inches	19–00¾
thickness, inch	00$\frac{31}{32}$
Number of furnaces, each boiler	1
furnace doors, each boiler	5
Grates, length, feet and inches	7–00
width, feet and inches	17–00
Total grate surface, square feet	1,428
Total heating surface, square feet	64,234
Ratio, G.S. to H.S.	44.98
Number of tube sections, each boiler	31
2-inch tubes, each boiler	1,100
4-inch tubes, each boiler	62
Distance between headers, feet and inches	8–00
Area of each smoke pipe, mean, square feet	101.56
G.S. ÷ area through smoke pipes	7.03
Kind of forced draft	Closed fireroom.
Number of oil burners, each boiler	8
Type of oil burners	Peabody.

FUEL—OIL SYSTEM.

In addition to the usual coal-burning appliances, a complete oil-burning system is provided. This installation is not intended for regular use, but only as auxiliary fuel to be burned in emergencies in combination with the coal.

The plant consists of two 10-inch × 7-inch × 12-inch, M. T. Davidson, heavy pressure, vertical, double-acting, single pumps,* one in each engine room on the center-line bulkhead. These pumps draw the fuel oil from the double bottom tanks, compartments C–90 to C–99 inclusive, located under the engine rooms and the space between the engine rooms and the firerooms, and deliver same to the oil burners on the boilers. Each boiler is fitted with eight oil burners, mechanically atomized, of the Peabody type.

A 4-inch suction leads from the bottom of each tank to the suction manifolds located near the pumps. There is a Macomb strainer in each pump suction and a special high-pressure strainer in each discharge. The discharge from each pump is 3½ inches, uniting into a common 3½-inch pipe (either pump being capable of supplying the entire system) which leads to the firerooms. This divides into two 2½-inch branches in the after fireroom, one supplying the burners on the port and the other on the starboard boilers. The port and starboard supply lines are symmetrically arranged and lead forward, outboard of the boilers, decreasing in size to 2 inches to the middle fireroom and 1½ inches beyond to the forward fireroom.

There are two oil heaters in each fireroom, one for the port and one for the starboard boilers, through which the oil is delivered to the burners in the same fireroom. The oil may be by-passed around the heaters if necessary. There are cut-out valves in the supply pipe to each pair of boilers, fitted with emergency deck-operating gear.

Four 6-inch filling connections are located on the main deck, which connect by two vertical pipes to the oil manifolds in the engine rooms.

Each tank is provided with a 1-inch steam connection for boiling out and the oil pumps are arranged for pumping the contents of the tanks overboard when washing them out, the latter connection being portable and only connected up when

* See also Table I.

cleaning the tanks, so there is no danger of flooding the system and tanks with sea water.

FORCED-DRAFT BLOWERS.

Four forced-draft blowers are installed for each fireroom. They are located in specially constructed blower rooms just below the protective deck and above the center of each fireroom.

The fans are of the Sirocco type built by the American Blower Company, and each is driven by a Diehl motor controlled from the fireroom working level or the blower room at will. Air is suppied from the fireroom ventilators, which are closed at the bottom when under forced draft. The fan data is as follows:

Revolutions per minute	695
Horsepower, each	40
Fan, diameter, inches	33
width, inches	30
Number of blades	64

*AUXILIARY FEED PUMPS.

There are three M. T. Davidson, 14½-inch × 9½-inch × 18-inch auxiliary feed pumps, one in each fireroom. They are of the vertical, double-acting, single type and are arranged so that any pump can feed any boiler.

*FIREROOM FIRE AND BILGE PUMPS.

In each fireroom there is an M. T. Davidson, 12-inch × 10-inch × 18-inch vertical, double-acting, single, fire and bilge pump, arranged to draw water from the bilge, the drainage system and the sea, and discharge to the fire main, sanitary system and overboard.

ASH HOISTS.

There are three Williamson Bros. ash-hoist engines of the two-cylinder, reversible type, one for each fireroom, located at the upper fireroom hatch. The port ventilator in each fire-

*See also Table I.

room contains the bucket guides, wire ropes, sheaves, etc. The hoists are operated from the main deck, at which deck the ash chutes are located at the ship's side.

The engines are of the following principal dimensions:

Number of cylinders	2
Diameter of cylinders, inches	4½
Stroke, inches	4½

ASH EXPELLERS.

In addition to the ash hoists, three Stone pneumatic ash expellers are fitted, one in each fireroom, discharging the ashes through the vessel's bottom.

The equipment in each fireroom consists of an expeller located just off the center line of the vessel, a 4-inch × 3-inch double engine, located overhead, which drives the former by means of shaft and gears. The ashes and clinkers, which have previously been broken up by striking with fire shovel, are fed through the hopper into the expeller, where they are crushed and delivered into a chamber in communication with the sea, from which they are discharged to the sea by air pressure. A Westinghouse, cross-compound air compressor, together with air reservoir and the necessary piping is provided for this purpose.

The air compressor data is as follows:

Number of compressers	3
Diameter of steam cylinders, inches	8½ and 14½
air cylinders, inches	9 and 14½
Stroke, inches	12
Number of air reservoirs	3
Capacity of each air reservoir, cubic inches	6,786
Working pressure, pounds	90

MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel. The two lines are cross-connected in the forward fireroom and in the engine rooms. The branches from the boilers are 5½ inches in diameter each

and the lines proper are 7½ inches in the forward fireroom, increasing to 9½, 11, 12½ and 14 inches at each successive boiler connection, the latter size being continued to the turbine-regulating valves in the engine rooms. No steam separators are fitted.

AUXILIARY STEAM PIPING.

From the main steam pipe in each engine room is a 7½-inch connection, which leads aft through the engine room and through the center-line bulkhead, forming a connecting loop between the two sides of the ship. From this pipe steam connections are taken for the various engine-room auxiliaries, steering engine and heating system aft.

In the firerooms the auxiliary steam consists of a 3-inch cross-connection between the port and starboard main steam lines, from which the branches to the auxiliaries are taken.

Steam connections for the dynamos, heating system and deck machinery forward are taken off the main steam cross-connection in the forward fireroom.

AUXILIARY EXHAUST PIPE.

An auxiliary exhaust pipe is fitted throughout the machinery spaces and elsewhere as required for the various auxiliaries. Connections are provided to direct the exhaust steam into either main or auxiliary condensers, either feed-water heater, or into the atmosphere through the after escape pipe at will. There are also connections for admitting the exhaust steam into the steam belts of the M.H.P. and L.P. ahead turbines when desired.

MAIN AND AUXILIARY FEED PIPES.

The main feed pumps in the engine rooms take their suction from the feed tank connecting pipe, the suction to each pump being 5 inches in diameter. The pumps discharge via the feed-water heaters, or by-pass same if desired, to the boilers. The discharge from each pair of pumps is 5 inches, uniting in the after fireroom into a 7-inch feed main leading

forward and diminishing in size as it goes forward to 6, 5 and 3½ inches.

The feed tank connecting pipe also has a nozzle for a 9-inch auxiliary feed suction main. This main gradually reduces in size as it leads forward to the auxiliary feed pumps. These pumps have direct connections to the boilers in their respective compartments, or can discharge into the main feed line. On each auxiliary feed pump is a grease extractor, which may be by-passed.

The branch main and auxiliary feed pipes to the boilers are each 2½ inches in diameter.

INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, time-fire device, telephones, voice tubes, etc., are fitted for transmitting orders and signalling to the various machinery compartments and other parts of the vessel.

AIR-COMPRESSOR PLANT.

Located in the engine room are nine 11-inch × 11-inch × 12-inch Westinghouse steam-driven air compressors and two air reservoirs of about 45,000 cubic inches capacity each, for use in running pneumatic tools in the engineering department, blowing soot off the boiler tubes and for the gas-ejecting system for the guns.

Each compressor has a capacity of about 360 cubic feet of free air per minute at 150 pounds pressure.

A pneumatic main, independent of the gun gas-ejecting system, is led throughout the machinery space, with branches to the workshop, evaporator and dynamo rooms, from which the connections for pneumatic tools and blowing soot off boiler tubes are taken.

EVAPORATING AND DISTILLING APPARATUS.

This plant is located on the berth deck, just forward of the engine hatches, with the distillers and evaporator feed-water heaters in the port engine hatch at high elevation. There are

four evaporators, four distillers and two feed-water heaters, with their accessories, arranged to operate in double effect. The plant has a combined capacity of 25,000 gallons of water per 24 hours.

For pumps see Table I.

Evaporator Data (each).

Type	Modified Bureau of Steam Engineering
Diameter, inside, feet and inches	5–9
Length over heads, feet and inches	6–3½
Tubes, number	112
diameter, inches	1½
thickness (brass)	Iron-pipe size.
Heating surface, square feet	311.24
Capacity, gallons per 24 hours	6,250
Diameter of steam connection (two), inches	2
vapor nozzle, inches	3
feed valve, inches	1¼
blow valves, inches	2

Distiller Data (each).

Type	Bureau of Steam Engineering.
Diameter, inside, inches	15½
Length between tube sheets, feet and inches	4–08½
Tubes, number	163
diameter, inch	00⅝
thickness, B.W.G.	16
Cooling surface, square feet	125.5
Diameter of circulating-water inlet and outlet, inches	03½
vapor inlet, inches	03
drain, inches	01½

Evaporator Feed-Water Heater (each).

Type	Alberger Condenser Co., triple flow.
Diameter, inside, inches	12
Length between tube sheets, feet and inches	3–09½
Tubes, number	126
size (corrugated copper), inch	00⅝
thickness, S.W.G.	20
Heating surface, square feet	60
Diameter of feed inlet and outlet, inches	02
vapor inlet and outlet, inches	03½

MACHINE SHOP.

A well-equipped machine shop is located amidships, between the engine hatches, at the berth deck level. Each machine tool is driven by its own motor.

The following machine tools are installed:

1 24-inch × 48-inch extension-gap lathe.
1 12-inch tool-room lathe.
1 14-inch tool-room lathe.
1 15-inch × 15-inch column shaper.
1 30-inch plain radial drill.
1 16-inch sensitive drill.
1 universal milling machine.
1 double emery grinder, wheels 12-inch diam. × 2-inch face.
1 portable cylinder-boring machine.

The tools are all up to date and complete, and are provided with the most modern attachments.

There is also a well-equipped blacksmith shop located on the main deck just abaft turret No. 3.

ELECTRIC PLANT.

There is one dynamo room located forward of the forward boiler room, and two distribution rooms, one forward and one aft, in which are located the lighting and power distribution boards for the respective parts of the ship. The forward distribution room also contains the generator boards.

The generator installation consists of four 6-pole, compound-wound, 300-kilowatt General Electric generators, each driven by a two-stage horizontal Curtis turbine. Each generator will deliver at normal load 2,400 ampères of current at 125 volts when running at 1,500 revolutions per minute. The generators are capable of delivering one-third overload for two hours without injury.

There are two condensers of elliptical section for the exclusive use of the dynamo turbines. Each condenser has its

independent air pump, centrifugal circulating pump and hot-well tank and pump. For pump data see Table I. The exhaust piping is so arranged with cut-out valves that either or both condensers can be used on any or all the generator sets.

The condensers are of the following principal dimensions:

Width, inside shell, feet and inches	4–00
Height, inside shell, feet and inches	5–08¾
Thickness of shell, inch	00½
Length, between tube sheets, feet and inches	6–08½
Thickness of tube sheets, inches	01
Number of tubes	2,190
Diameter of tubes, inch	00⅝
Thickness of tubes, B. W. G	16
Diameter exhaust nozzle, inches	28
air-pump suction, inches	07
circulating water inlet and outlet, inches	09
Cooling surface, square feet	2,403

TORSION METER.

Each line of shafting is fitted with a Gary-Cummings torsion meter for ascertaining the shaft horsepower of the main turbines.

As a complete description of this instrument, by Commander U. T. Holmes, U. S. N., appears elsewhere in this number, further remarks on its construction and operation are unnecessary here, except that it gave satisfactory and consistent results on the trials.

TRIALS.

Standardization Trial.—This trial was run on the Rockland, Maine, course Monday, March 25, 1912. The weather was clear with a brisk wind blowing almost parallel to the course from the north. The first run was commenced about 6:30 A.M. in a southerly direction, the runs being alternately south and north. In all twenty-two runs were made over the measured mile at various speeds, but the data from only twenty runs was used in compiling the speed and power curves, runs

TABLE I.—PUMPS AND PUMP CONNECTIONS.

No.	Pumps.	Size (inches).	Kind.	Suction pipes from—		Discharge pipes to—		Location.
				Ins.		*Ins.*		
2	Main air.....	2 17½ by 2 35 by 21..	Twin, vertical, bucket, single-acting.	12	Condenser.....	10	Feed tank.....	One in each engine room.
				4½	Turbine drains.			
				12	Augmenter.			
2	Main circulating.	11½ by 23 by 12.....	Vertical, compound, centrifugal.	2–21	Sea.....	30	Condenser.....	One in each engine room.
				15½	Main drain.	30	Overboard.	
						6	Augmenter.	
4	Main feed.....	14½ by 9½ by 18.....	Vertical, piston, double-acting, single.	5	Feed tank connecting pipe..	4	Main feed discharge	Two in each engine room.
				5	Channel way cross-connection pipe.			
3	Auxiliary feed..	14½ by 9½ by 18.....	Vertical, piston, double-acting, single.	5	Feed-suction pipe.....	4	Auxiliary feed main.....	One in each boiler room.
				3½	C. and R. pipes from ship's sides.a	3	Overboard.	
						3½	Reserve feed tanks.a	
				3½	Reserve feed tanks.			
				2½	Hose connection.			
4	Engine room, fire and bilge.	12 by 10 by 18.....	Vertical, piston, double-acting, single.	5	Sea.....	5	Fire main.....	Two in each engine room.
				5	Engine-room bilge.	5	Overboard.	
				4½	Shaft alley.	5	Sanitary.	
				5	C. and R. drainage manifolds.	3½	Oil cooler.	
						2½	Hose connection.	
				1½	Hose connection.			
3	Fireroom, fire and bilge.	12 by 10 by 18.....	Vertical, piston, double-acting, single.	5	Fireroom floors.....	5	Fire main.....	One in each boiler room.
				5	Sea.	5	Overboard.	
				5	C. and R. drainage manifolds.	5	Sanitary.	
						2½	Hose connection.	
				1½	Hose connection.			

2	Oil cooler, circulating.	8 by 7 by 8	Vertical, piston, double-acting, single.	4	Sea	3½	Oil cooler	One in each engine room.
2	Fuel-oil supply	10 by 7 by 12	Vertical, piston, double-acting, single.	4	Storage tanks	3½ 2½	Boiler room Overboard.	One in each engine room.
4	Forced lubrication service.	10 by 9 by 12	Vertical, piston, double-acting, single.	5	Lubricating-oil tanks	4½	Cooler	Two in each engine room.
2	Dynamo-room hot well.	4½ by 5 by 6	Vertical, piston, double-acting, single.	2½	Hot-well tank	2	Main feed tank cross-connecting pipe.	In dynamo room.
2	Distiller circulating.	12 by 14 by 16	Horizontal, piston, double-acting, single.	6½	Sea	6	Distillers and fire main	Evaporator room.
2	Auxiliary air and circulating.	8 by 10 by 12 by 12	Combined, hor. piston, double-acting, single.	6 6	Auxiliary condenser Sea.	4½ 6	Main feed tanks Overboard.	One in each engine room.
2	Dynamo condenser air.	9 by 18 by 12	Vertical, bucket, single-acting.	7	Dynamo condenser	6	Dynamo-room hot well	In dynamo room.
2	Dynamo condenser circulating.	6 by 6	Centrifugal	9	Sea	9	Dynamo condenser	In dynamo room.
2	Evaporator feed.	4½ by 5 by 6	Vertical, piston, light-service, double-acting, single.	2½	Sea	2	Evaporators and feed heaters.	Evaporator room.
1	Distiller fresh water.	4½ by 5 by 6	Vertical, piston, light service.	2½ 2½	Bottom of distillers Test tank.	2 2 2	Fresh-water tanks Main feed tanks. Reserve feed-water tanks.	Evaporator room.
2	Pipe insulator circulating.	6 by 8 by 8	Vertical, piston, double-acting, single.	3½	Sea	3	Pipe insulators	One in each engine room.
1	Reserve feed	4½ by 5 by 6	Do	2½	Reserve feed tanks	2	Main feed tanks	Port engine room.

a No. 2 boiler room only.

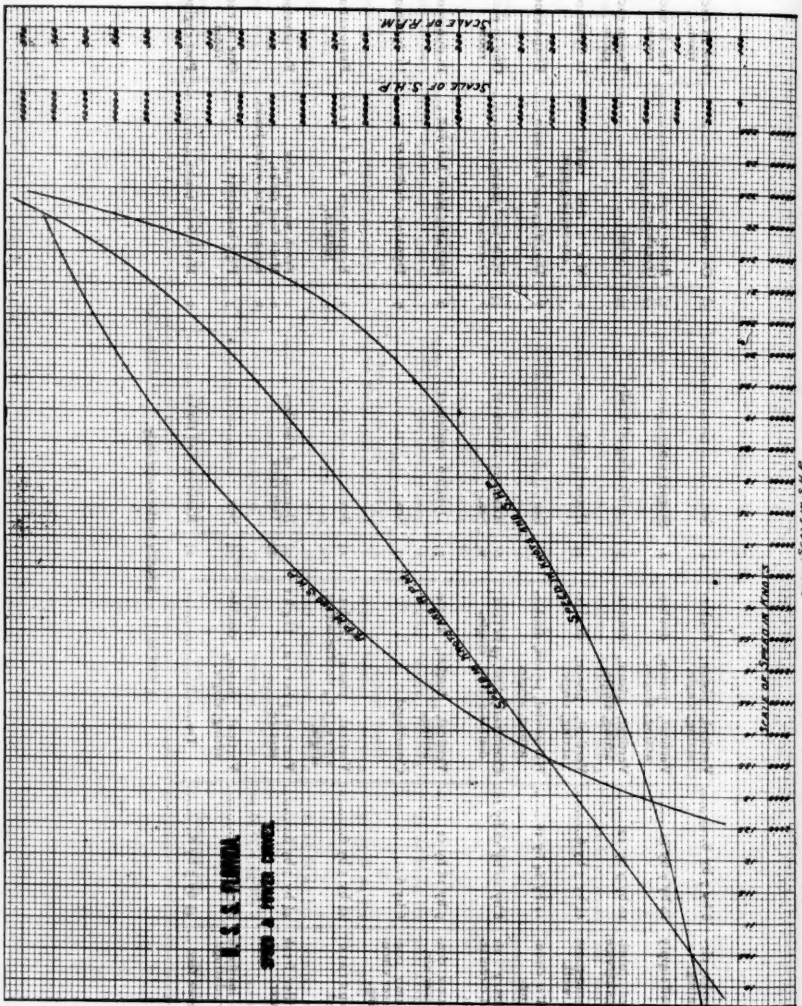

SCALE OF R.P.M.
SCALE OF S.H.P.
SCALE OF SPEED IN KNOTS
SCALE OF SHIP'S H.P.
SPEED AND H.P. CURVES
U. S. S. FLORIDA
SPEED H.P. CURVES

TABLE II.—U. S. S. "FLORIDA."—STANDARDIZATION TRIAL DATA.

Number of run	1	2	3	4	5	6	7	8	9	10	11	12	15	16	17	18	19	20	21	22
Direction of run	S.	N.	S.	N.	S.	N.	S.	N.	S.	N.	S.	N.	S.	N.	S.	N.	S.	N.	S.	N.
Time on course, minutes and seconds	4-9.1	5-20.9	4-11.3	6-38.1	5-21.8	7-12.4	3-31.4	4-10.1	3-36	3-18.8	3-4.7	3-17.9	2-41.6	2-49 5	2-39.7	2-45.9	2-40.8	2-57.8	2-49.7	3-0.7
Revolutions of main engines, mean:																				
To make one knot	798.23	996.57	802.23	994.58	787.08	1,020.68	823.95	958.63	849.65	936.75	872.75	931.68	961.20	1,003.90	965 28	991.45	960 75	946.50	904.30	954 25
Per minute	192.30	186.39	191.54	149.88	146.77	141.71	233.80	230.13	236.11	282.79	283 74	282.61	356.87	355 57	362.65	358 35	358.59	319.31	319.83	316.93
Speed, in knots	14.452	11.218	14.325	9.043	11.187	8.326	17.029	14.394	16.667	18.109	19.491	18.191	22.277	21.239	22.542	21.700	22.388	20.247	21.214	19.923
Horsepower:																				
Main engines	5,446	5,248	5,356	2,755	2,460	2,389	9,742	9,872	10,108	18,279	18,271	18,313	36,078	38,003	39,251	38,615	37,984	26,471	25,885	25,673
Auxiliaries	670	670	652	685	637	603	755	746	723	872	859	875	1,210	1,261	1,252	1,229	1,228	1,153	1,101	1,122
Total	6,116	5,918	6,008	3,440	3,097	2,992	10,497	10,618	10,831	19,151	19,130	19,188	37,288	39,264	40,503	39,844	39,212	27,624	26,986	26,795
Steam pressure, mean:																				
Main steam, gage	207.5	208.5	201.5	212.5	212.5	206.5	206.5	217	208.5	199	208.5	205.5	186	184.5	187	181	175.5	186	185	185
At M.H.P. turbines, absolute	18.5	17.5	19	13.25	12	12	40	37.5	41	67.5	67.5	67.5	156	160	166.5	162.5	165.5	141.5	113.5	112.5
At H.P.C. turbine, absolute	125	125	127	80	73	72	5	9	12	10.5	8	7.5	12.5	8	8	10	10	10	8	8
At I.P.C. turbine, absolute	50	55	58	35	31	30	115	110	116	188	189	188	13.75	13.75	13.75	13.75	13.75	13.75	13.5	13.5
At L.P. turbines, absolute	6	6	5.75	5.25	5 1	5.1	7.75	7.6	8	9	10.25	10.25	16	17.5	18.25	17.25	17.75	15.88	13 6	14.1
Vacuum, in inches, mean	29.1	29.15	29.15	29.2	29.25	29.25	28.95	29.1	29.05	28.8	28.85	28.8	28.05	28	27 95	27.95	27.9	28.15	28.45	28.45
Number of turbines in operation	Six.						Five.							Four.						

TABLE III.—U. S. S. "FLORIDA."—TRIAL DATA.

.....	4-hour full-power trial.	24-hour, 19-knot endurance trial.			24-hour, 12-knot endurance trial.
Speed, in knots	22.08	19.19			12.08
Slip of propellers, per cent. of own speed, mean	27.67	20.98			19.32
Draught, mean on trial, feet and inches	27-10	27-09$\frac{1}{32}$			27-06$\frac{1}{32}$
Displacement, corresponding, tons	21,240.	21,215.			20,976.
Number of turbines used	4.	Five, first 5 hours; four, last 19 hours.			6.
boilers used	12.	12.			8.
Grate surface used, square feet	1,428.	1,428.			952.
Heating surface used, square feet	64,236.	64,236.			42,824.
Pressures (average):					
Main steam, at boilers, pounds gage	210.	202.			203.
engine rooms, pounds gage	179.2	189.1			197.4
		(*a*)		(*c*)	
In steam belt, H.P.C. turbine, pounds absolute	10.8	9.3		9.2	114.4
I.P.C. turbine, pounds absolute	13.7	178.		12.2	49.7
M.H.P. turbines, pounds absolute	173.1	73.6		87.6	17.2
L.P. turbines, pounds absolute	18.5	10.		11.5	5.
Barometer, inches of mercury	30.2	30.17		30.14	30.04
Vacuum in condensers, inches of mercury	28.4	28.9		28.5	29.6
Main feed line, at pumps, pounds gage	278.	268.7			265.8
Auxiliary exhaust in feed heater, pounds gage	6.7	4.9			6.4
Forced-lubrication system, pounds gage	13.3	12.6			11.1
Air pressure in firerooms, inches of water	1.99	0.6			0.
Temperatures, degrees *F.* (average):					
Main injection	39.8	52.			45.
overboard discharge	64.1	67.7			52.8
Feed water	197.8	201.			223.3
Engine rooms, working platform	81.8	98.5			94.6
Firerooms, working level	79.	98.			100.
Outside air	31.	53.2			53.4
Smoke pipes	620.	536.			381.
Revolutions or double strokes per minute (average):		(*a*)	(*b*)	(*c*)	
Shaft No. 1	366.64	275.40	289.59	293.32	165.65
No. 2	361.24	319.36	293.68	286.93	195.45
No. 3	363.62	272.72	285.43	288.77	188.66
No. 4	364.11	274.26	289.32	293.29	164.29
Mean, all shafts	363.90	285.44	289.50	290.58	178.51
Main air pumps	23.5	18.8			17.3
circulating pumps	244.3	201.3			161.8
feed pumps	26.3	20.6			16.8
Forced-draft blowers	712.	573.			Not used.

		(*a*)	(*b*)	(*c*)	
Horsepower (average):					
Shaft No. 1, S.H.P.	10,039.	3,766.	4,488.	4,640.	646.
No. 2, S.H.P.	10,370.	7,531.	5,281.	4,807.	1,583.
No. 3, S.H.P.	10,148.	3,932.	4,710.	4,874.	1,415.
No. 4, S.H.P.	9,954.	4,000.	4,880.	5,065.	789.
Total S.H.P., all shafts	40,511.	19,229.	19,359.	19,386.	4,433.
I.H.P., main air pumps	29.		21.		10.
circulating pumps	507.		315.		167.
feed pumps	129.		99.		41.
chargeable to forced-draft blowers	333.		178.		Not used.
other auxiliaries	301.		250.		246.
Total I.H.P., all auxiliaries	1,299.		863.		464.
horsepower, all machinery	41,810.	20,092.	20,222.	20,249.	4,897.
Water-consumption data:		(*a*)	(*b*)	(*c*)	
Pounds per hour, main engines	493,428.	244,901.	251,669.	253,450.	74,543.
auxiliaries	67,352.		56,382.		41,843.
all machinery	560,880.	301,283.	308,051.	309,832.	116,386.
main engines, per S.H.P.	12.180	12.736	13.	13.074	16.815
auxiliaries, per I.H.P.	51.849		65.333		90.179
all machinery, per total H.P.	13.415	14.995	15.233	15.301	23.767
S.H.P.	13.845	15.668	15.913	15.982	26.254
Coal-consumption data:					
Kind and quality	George's Creek, very good.		Good.		Good.
B.T.U. per pound	No sample taken.		No sample taken.		No sample taken.
Pounds per hour	66,586.		34,720.		10,920.
per S.H.P.	1.644		1.793		2.463
total H.P.	1.593		1.717		2.23
square foot of grate	46.629		24.314		11.471
heating surface	1.037		.541		.955
Miscellaneous:					
S.H.P. per square foot of grate	28.369		13.557		4.657
Square feet of heating surface per S.H.P.	1.586		3.318		9 660
cooling surface (main condensers) per S.H.P.	0.752		1.574		6.873
Pounds of water evaporated per pound of coal per hour	8 423		8.872		10 658
sq. ft. of H. S. per hour	8,732		4.796		2.718
Knots per ton of coal	0.743		1.238		2.478

(*a*) Averages of first five hours of trial, with five-turbine combination.
(*b*) Averages of entire trial.
(*c*) Averages of last nineteen hours of trial, with four-turbine combination.

Nos. 13 and 14, not being up to the speed desired, were thrown out.

From the data obtained it was found to require 321.4 r.p.m. of the main engines to attain the designed speed of 20.75 knots, 286.3 r.p.m. for 19 knots, and 177.2 r.p.m. for 12 knots.

Table II contains the data obtained on the various runs, from which the curves, Plate I, were plotted.

Four-Hour Full-Speed Trial.—This trial took place at sea off Rockland, Maine, on Tuesday afternoon, March 26, 1912. The weather was clear with gentle breezes and a smooth sea. The designed speed was easily exceeded, an average speed of 22.08 knots being attained, and the designed water rate was likewise bettered.

Table III gives a comparative list of the data obtained on this and the following trials.

Twenty-Four Hour 19-Knot Endurance and Coal and Water-Consumption Trial.—The trial commenced at 1:20 A. M., March 27, 1912, and terminated at the same hour on the following day. The weather was clear with light airs and a smooth sea. Unfortunately, unsatisfactory dummy clearances developed in the starboard L.P. turbine, which rendered it necessary to disconnect the I.P.C. turbine after five hours' running, and to continue the trial with the four-turbine combination, but in spite of this handicap the water consumption was within the designed rate.

For data see Table III.

Twenty-Four Hour 12-Knot Endurance and Coal and Water-Consumption Trial.—This trial was run on the six-turbine combination, March 28 and 29, 1912, the start being at 4:20 A. M. The weather was excellent throughout the trial, there being no wind and a smooth sea.

The water consumption on this trial was a little in excess of the designed rate, due to the high consumption of the auxiliaries.

The data obtained on this trial is given in Table III.

THE GARY-CUMMINGS TORSIONMETER.

By Commander U. T. Holmes, U. S. N., Member.

The accompanying figure shows this instrument as supplied to the U. S. S. *Florida*. A tube *A*, about fourteen feet in length, is installed inside a section of the propeller shafting. One end of this tube has clamped on it a spider casting *B*, as shown at the left and in section at *X*–*X*. Each leg of this spider carries a spindle *C*, which is a close fit in *B*, with one end turned to a sharp point and the other cut to form a feather that fits in a tapered keyway on screw *D*. The three keyways in *D* are about $\frac{3}{16}$ inch deep at the left end of screw and taper to about $\frac{1}{32}$ inch at the right. A long-handled socket wrench fits nut *E* and is actuated from the left end of shaft. The screw plug *D* is thus drawn to the left, and, since the keyways in which spindles *C* fit are tapered, the points on *C* bite into the shaft, thus binding the spider and the left end of tube rigidly to the inner wall. *F* is a lock washer which serves to prevent *E* working loose.

The right end of tube *A* has a casting *G* clamped on it, and this rests in a ball bearing fitted into a spider set into the end of shaft and held firmly by means of tapered screws *H*.

The arrangement described above permits the tube *A* to remain free from torsion while the shaft is transmitting power. The left end is clamped to shaft, but the right end is permitted to turn freely in its ball bearing and will do so in case there should be an angular displacement of the shaft at this end. It is evident that there will be such a movement and the amount will be proportional to the twisting moment, or torque, transmitted by the shaft.

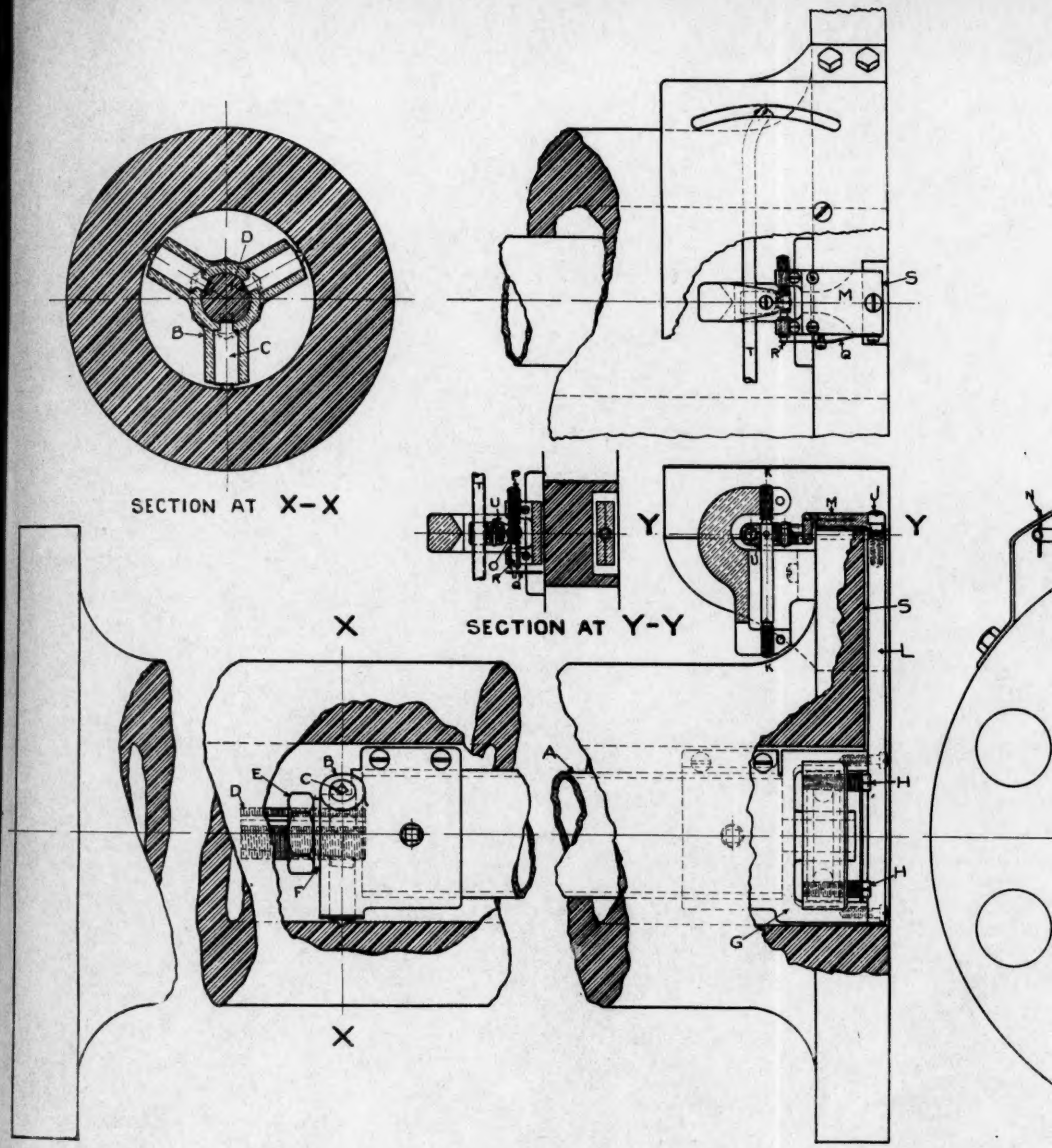
It remains, then, to provide a means for measuring and recording this movement. A lever *L* is secured to the right

end of tube *A* and lies in the slot *S*, cut for this purpose in the face of flange on the shaft. The width of this slot is such as to freely permit the small movement of lever caused by the slight angular movement of tube relative to shaft. A top piece *M* secured to *L* by screw *J* simply serves to transmit the movement of *L* to the recording mechanism. A double arm *T*, made of tubing to give the maximum strength on minimum weight, is fitted in a block *U* that is centered at *KK* and actuated by *M*. At each end of *I* is a german-silver recording needle, secured by binding screw not shown in figure. The ball *O*, section *Y–Y*, is held against screw *P* by spring *Q* and pin *R*, thus providing for taking up all lost motion due to wear.

Since *I* and *O* are both firmly secured to the block *U* moving about *K–K* as a center, any movement of *L* with respect to the flange on shaft causes one of the two points *NN* to go to the left and the other to the right. It still remains to provide means to record this movement, that is, the distance between the circular paths traced by the two points *NN* as they revolve with the shaft.

At the lower right-hand corner of the figure is shown the device for holding the small sheets of metal-coated paper for taking the record from the needles *NN* as they revolve in the paths *V–V*. Several sheets of this paper are placed loosely in the pocket provided, and by pushing the rod *W* the top sheet is moved until it becomes tangent to *V–V* and receives the record. *Z* is a handle for moving the paper holder so that if it should be found that only one needle touches, the plane of the paper can be slightly changed so that both needles will mark. *S'* is an adjusting screw for moving paper holder in and out from center of shaft.

The linear distance between the lines traced by *NN* is a direct measure of the twist of shaft when corrected for the zero reading. This is obtained by dragging the propeller in order to cause the shaft to revolve when transmitting no power. Both points can be made to mark the same line by careful adjustment of the screw *P*, but this is inadvisable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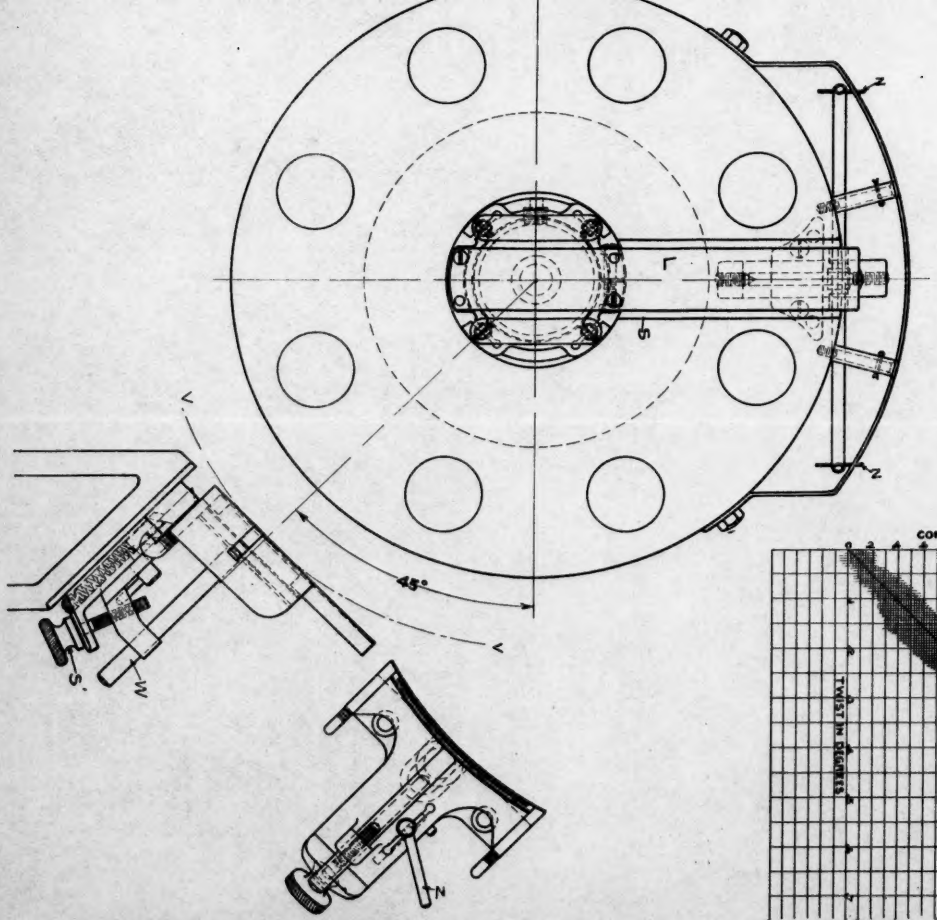
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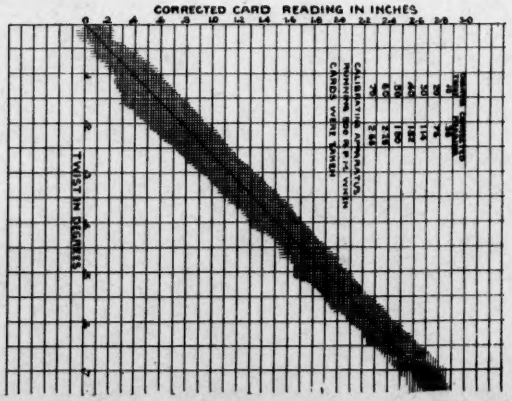

CORRECTED CARD READING IN INCHES
TWIST IN DEGREES

since the shaft must be stopped to make the adjustment and a zero reading of as much as 0.1 inch is of no disadvantage.

The proportions of the *Florida's* instruments are such that one-half degree angular displacement of shaft is equivalent to 1.9 inches of displacement between the two recording points. This ratio was found in shop tests with a calibrating instrument especially devised for the purpose. The results of these tests are plotted in the upper right-hand corner of the figure, and show that the ratio between angular displacement and distance between lines on the record is constant for all powers. The tests were made first by adjusting the arm in the slot, using a micrometer gage to place it in the exact position required. The arm was then locked in this position and the calibrating instrument run in a lathe and cards taken while running.

It was also shown by these experiments that the speed of rotation does not affect the reading of the instrument. Centrifugal force only tends to increase the diameters of the circles traced by the needles, and there is no tendency to vary the distance between the planes of these circles. The linear distance between marks on a card when laid flat is directly proportional to the angular twist of shaft. It is thus seen that one constant serves for all powers and speeds.

To obtain the horsepower constant the shafts are calibrated in the shop by applying a known moment and measuring the angle of twist in a known length by means of battens. Thus in determining the horsepower constant for No. 2 shaft on the *Florida* the following method was employed:

From calibration data it was found that one-half degree of twist in 100 inches of length of shaft corresponds to 171,371 foot pounds of moment. One inch reading on torsionmeter card corresponds to 0.263 degree of twist. Hence 0.263 degree of twist when the shaft is transmitting power must correspond to $0.263 \times 171{,}371 \times 2 = 90{,}312$ foot pounds of twisting moment or torque.

$$\text{S.H.P.} = \frac{2\pi \times M \times \text{R.P.M.}}{33{,}000},$$

where M = twisting moment in foot pounds. If the torsionmeter reading is one inch we could substitute for M the number 90,312, but the reading varies directly with the torque in all cases, and

$$\text{S.H.P} = \frac{2\pi \times (90{,}312 \times T) \times \text{R.P.M.}}{33{,}000} = 17.195 \times T \times \text{R.P.M.},$$

where T = the corrected torsionmeter reading in inches.

The above expression applies to a length of 100 inches in No. 2 shaft of the *Florida*. In any shaft the constant is inversely proportional to the length of shaft under consideration. It is therefore possible to cut the tube A of such length that the constant will have a value of 10, and the work of calculating horsepower from the torsionmeter reading can be thus greatly facilitated.

This instrument possesses the advantage of being housed in the shaft where it is entirely out of the way. There are few parts to get out of order, there is no chance for lost motion to develop and the weight is less than in other torsionmeters. It is applicable only to hollow shafting. The tube and attachments must be placed there permanently and cannot be adjusted or even examined without removing the section of shaft in which installed. The parts are rigidly constructed with locking devices on all pieces that could otherwise work loose. When carefully installed it is believed that there will be no necessity for further adjustment of these parts.

FORMULAE.

The moment M, caused by the pull W on an arm of length $r = W \times r$. From the results of calibration experiments a curve is constructed, the ordinates giving the values of M in *foot pounds* and the abscissae the corresponding readings of the torsionmeter. From this curve the moment corresponding to any reading of the torsionmeter, when the shaft is transmitting power may be obtained.

The shaft horsepower (S.H.P.) is equal to

$$(1) \qquad \frac{2 \times \pi \times M \times \text{R.P.M.}}{33{,}000} = \frac{M \times \text{R.P.M.}}{5{,}252}.$$

From applied mechanics we know that the angle of torsion, for a shaft within the elastic limit,

$$(2) \qquad \varphi = \frac{2 \times m \times l}{E \times I \times d} = \frac{\Delta}{r} = \frac{\pi\theta}{180}, \text{ where}$$

φ = the angle of torsion in circular measure;
θ = the angle of torsion in degrees;
Δ = the linear deflection measured on a circumference of radius r;
m = the twisting moment in *inch pounds*;
l = the length of shaft in inches;
d = the outside diameter of the shaft in inches;
d_1 = the inside diameter of the shaft (if hollow) in inches.
E = the modulus of elasticity of the shaft;
I = the modulus of the section $= \frac{\pi d^3}{16}$ for solid shafts, or $\frac{\pi}{16}\frac{(d^4 - d_1^4)}{d}$ for hollow shafts.

It is more convenient to express the twisting moment in foot pounds, $M = \frac{m}{12}$. Making this change, we have

$$(3) \quad M = \frac{E \times I \times d \times \pi \times \theta}{360 \times 12 \times l} = \frac{\pi^2}{16} \times \frac{E \times d^4}{360 \times 12 \times l} \times \theta.$$

Substituting this value of M in formula (1) for horsepower, we have

$$(4) \quad \text{S.H.P.} = \frac{2 \times \pi \times M \times \text{R.P.M.}}{33{,}000} = \frac{2 \times \pi^3 \times E \times d^4 \times \theta \times \text{R.P.M.}}{360 \times 16 \times 12 \times 33{,}000 \times l}$$
$$= \frac{E}{36{,}782{,}000} \times \frac{d^4 \times \theta \times \text{R.P.M.}}{l} \text{ for solid shafts.}$$

Similarly for hollow shafts,

$$(5) \quad \text{S.H.P.} = \frac{E}{36{,}782{,}000} \times \frac{(d^4 - d_1^4) \times \theta \times \text{R.P.M.}}{l}.$$

From (3) we have

$$(6)\quad E = \frac{360 \times 12 \times 16 \times l \times M}{\pi^2 \times d^4 \times \theta} = \frac{7{,}003.2 \times l \times M}{d^4 \times \theta}$$

$$= \frac{583.6 \times l \times m}{d^4 \times \theta} \text{ for solid shafts;}$$

and similarly,

$$(7)\quad E = \frac{7{,}003.2 \times l \times M}{(d^4 - d_1^4) \times \theta} = \frac{583.6 \times l \times m}{(d^4 - d_1^4) \times \theta} \text{ for hollow shafts.}$$

From (6) or (7) the modulus may be calculated when the results of calibration of the shafting have been obtained. This has been done for the shafting of a large number of our vessels, and the results are given in the following table:

Ship.	Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Shaft No. 4.	Outside diam., inches.	Inside diam., ins.
North Dakota	11,495,800	11,590,800	.....	.....	14	8
Preston	11,825,555	11,926,444	11,825,555	.....	6½	3¼
Flusser	11,548,000	12,100,000	11,730,000	.....	6½	3¼
Reid.....	11,500,000	12,030,000	11,810,000	.....	6½	3¼
Monaghan	11,550,000	11,675,000	11,360,600	.....	6¼	3¼
Terry.....	11,219,000	11,396,000	11,505,000	.....	6¼	3¼
Chester	11,622,702	11,820,000	11,840,000	11,755,405	8 (solid)	
Salem.....	11,373,947	11,687,837	.....	.....	9¼	5
Perkins	11,490,000	11,551,000	.....	.....	8	4
Sterett	11,484,000	11,635,000	.....	.....	8	4
McCall	11,685,887	11,891,530	11,825,900	.....	6¼	3¼
Burrows	11,952,038	11,895,519	11,878,224	.....	6¼	3¼
Florida	11,695,300	11,659,600	11,616,600	11,844,600	12½	6
Arkansas	11,936,903	11,962,910	11,893,953	11,818,720	12½	8
Test shaft	11,577,071	.....	.....	.....	1¼ (solid)	

These results do not show great variation, but the difference is sufficient to justify the rule that all shafting on which a torsionmeter is to be used must be calibrated. It is probable that the results would have approximated more closely to a mean value if the axial hole had in all cases been exactly central and of the correct diameter throughout, and also if the experiments had been carried out under the same conditions by the same observers.

When calibration is impracticable, approximately correct

results in the case of high-grade steel shafting will be obtained if we assume a value for E of 11,750,000. Substituting this value in (4) and (5) we have

$$(8) \quad \text{S.H.P.} = \frac{d^4 \times \theta \times \text{R.P.M.}}{3.13 \times l} \text{ for solid shafts, and}$$

$$(9) \quad \text{S.H.P.} = \frac{(d^4 - d_1^4) \times \theta \times \text{R.P.M.}}{3.13 \times l} \text{ for hollow shafts.}$$

SCREW PROPELLERS.

DETERMINATION OF DIAMETER, PITCH AND PROJECTED AREA BY MEANS OF THE EFFECTIVE THRUST.

By Captain C. W. Dyson, U. S. N., Member.

In previous work which the writer has submitted on this subject, the assumption was made that the apparent slips and propulsive coefficients of propellers working in the wakes of similar vessels, at equal indicated thrusts per square inch of projected area, varied directly as the projected-area ratios. That is, if we laid down the slips and the propulsive coefficients for a propeller of .32 projected-area ratio and those for a .54 projected-area ratio, both working in the wakes of vessels of standard forward and after body form and of the same standard block coefficient, the slips and propulsive coefficients for propellers of other projected-area ratios would, for any value of indicated thrust per square inch of projected area, lie on a straight line connecting the points of this thrust value on the .32 and .54 projected-area ratio curves.

This supposition is not considered satisfactory, as it tends to aggravate errors which must necessarily exist in work of this kind.

Again, the projected-area ratios used in the construction of the previous charts are so close together that any error in either, and particularly in the .54, for the propulsive coefficients magnifies the farther we depart from the .32 towards the larger values of projected-area ratios, so that if the propulsive coefficient given for the .54 is too high, all the propulsive coefficients for ratios greater than .32 will be too high and the error will be much increased when we reach projected-area ratios in the .60's. This would result in making all propellers of projected-area ratios greater than .32, for which

ratio the propulsive coefficients are checked by the performances of two vessels and are considered as accurate as it is possible to obtain, too small for the proper absorption of the designed power.

Since the work referred to has been published, carefully observed trials have been run with propellers of much greater projected-area ratios than .54, and it is considered advisable to revise the charts, using projected-area ratios of .32 and of .624, the latter being taken from the trials of a three-shaft torpedo-boat destroyer, using the data of performance of the wing screws only. The result of this choice will be to make the propulsive coefficients for the larger ratios slightly less than can be realized, thus giving slightly larger propellers than are absolutely necessary for trial conditions. This is a fault in the right direction, as such screws will be all the better when the vessels are loaded down to cruising draught.

TABULATION OF CHARTS.

The charts submitted and which will be described in order, are—

Plate 1. Corrective chart for Block Coefficients where greatest immersed beam divided by length on load water line varies from that of the standard vessels from which the slip charts are derived.

Plate 2. Curves of Appendages resistance in percentage of bare hull resistance.

Plate 3. Curves of Propulsive Coefficients.

Plate 4. Curves of Tip Speeds.

Plate 5. Curves of Propulsive Coefficients, Tip Speeds, *I. T.*'s, *P. T.*'s and *E. T.*'s for points of design.

Plate 6. Curves of Apparent Slips for points of design.

Plates 5 and 6 should not be used for *P. A.* ratios less than .2.

Plate 7. Curves of Propulsive Coefficients, Tip Speeds, *I. T.*'s, *P. T.*'s and *E. T.*'s for $\frac{P.A.}{D.A.} = .2$. To be used when allowed Tip Speeds are less than 5,000 feet per minute.

Plate 8. Standard Projected-Area Forms, and Curves showing Helicoidal (developed) Area ratios corresponding to different values of projected-area ratios for varying values of ratio between Pitch and Diameter.

DESCRIPTION OF CHARTS.

Plate 1. Corrective Chart for Block Coefficients.—This chart has already been described in the February number of this journal, but in order to make the present article as complete as possible, a short description of it will be given:

X is the line of ratios of $\frac{B}{L.\ W.\ L.}$ for the parent ships upon whose performances the preliminary slip charts are based. This line is marked standard $\frac{B}{L.\ W.\ L.}$, and these ratios are standard for the values of block coefficients given as abscissæ. The curve above marked standard coefficient of immersed midship section is to be used in conjunction with this line X, in the determination of the relative fineness of the fore and after bodies of any vessel under consideration as compared with the standard vessel.

Should the vessel plot for $\frac{B}{L.\ W.\ L.}$ directly on X and the coefficient of immersion of midship section directly or nearly on the standard curve, the curves of effective thrust for the block coefficient indicated by the plotted points should be used provided the vessel is twin screw.

Should the coefficient of immersion of midship section be larger than the standard, the vessel will be finer, or, if smaller than the standard, fuller than the parent ship, and the block coefficient to use in selecting the effective thrusts must be modified accordingly, the judgment of the designer must be used as to the degree of change from the standard.

Should the vessel plot above or below C as at B and at A, draw a line through the plotted point and the abscissa value unity. Where this line crosses X will give the block coefficient value to use in selecting the effective thrust, should the

Errata.

Page 576:

Fourth line from bottom, for "C, B and A", read "X, E and A".

Page 619, Problem 2:

Fifteenth line of data, "s" should be "s_1";

Sixteenth line of data, "S_1" should be "S";

Twentieth line of data, "$T.\ S.$" should be "$T.\ S_1$".

standard coefficient of immersion of midship section corresponding to this point of intersection agree with the coefficient of immersion of midship section in question. Should it differ, correction should be made as before.

Should the vessel be fitted with four shafts, have twin screws carried by a Lundborg stern, or have a center screw with the dead wood cut away to give a free flow of water to the screw, the inboard screws in the first case, the twin screws in the second, and the center or single screw in the third, will feel the effect of the hull much more than will twin screws carried by struts. The wake in which they work will be heavier and a correction for block coefficient must be made accordingly.

The same effect, in a greater degree, will occur with a single screw or with the center screw of a three-shaft arrangement, where the screws work in the wake of and close to the stern post. The block coefficient lines to use for the location correctives are marked respectively Y and Z. The fineness as compared with the standard vessels is checked by comparing the coefficient of immersed midship section corresponding to the intersection of the corrective line through the plotted point A or B of the vessel and abscissa value unity with the line X, but the block coefficient correction which we judge should be made due to difference in fineness from the standard, will be applied to the block coefficient abscissa value corresponding to the point of intersection of the corrective line with Y or Z as the case may be. Should the fineness be standard, the block coefficient value as indicated by the abscissa value of this latter intersection will be used.

Plate 2. Curves of Appendage Resistance.—In earlier work on this subject of appendages the writer gave a table of percentage resistances due to each kind of appendage usually fitted to the hull of a vessel. It is a much readier and better way to plot the percentage appendage resistance on $\frac{B}{L.W.L.}$ as abscissæ and draw fair curves through the plotted points. From such curves can readily be seen the influence of depart-

ure from standard fineness of hull upon the percentage resistance. The curve marked standard is the one usually to be used except in cases where great care is used in fitting the appendages, when the curve directly below the standard should be used, or where struts are much reduced in size, as in Parsons four-shaft ships, shown by curve marked for such vessels or where the bare hull resistance is abnormally large, as shown by curve marked *Massachusetts* and *Nebraska*.

The curve showing the influence of large scoops has been checked by trials of the model of destroyers 17 to 21, fitted with large scoops, and *has been found to agree almost exactly*.

CONSTRUCTION OF CHARTS OF DESIGN.

Curves of Propulsive Thrust.

In laying down these thrusts which are represented by the following formula:

$$P.T. = \frac{I.H.P. \times 33{,}000 \times \text{Propulsive Coefficient}}{\text{Pitch} \times \text{Revolutions} \times \text{Proj. Area of Propeller in sq. ins.}}$$

the performances and propulsive coefficients of a propeller of .32 projected-area ratio and one of .624 projected-area ratio, the forms of the blade projections in both cases being standard, were used.

Having obtained for these two propellers the indicated thrust per square inch of disc area and of projected area, by using the following formulas:

$$I.T. \text{ per sq. in. Disc Area} = \frac{I.H.P. \times 33{,}000}{P \times R \times \text{Dis. Area in square inches}};$$

$$I.T. \text{ per sq. in. Proj. Area} = \frac{I.H.P. \times 33{,}000}{P \times R \times \text{Proj. Area in square inches}}$$

and the propulsive thrust for each, the propulsive thrust values were laid down as curves and marked .32 and .624. The ordinates of these curves are pounds propulsive thrust per square inch of projected area, and the abscissæ are pounds indicated thrust per square inch of disc area.

In interpolating for other values of $\frac{P.A.}{D.A.}$, the following hypothesis was made:

All screw propellers may be considered as lying between the following two limits: (*a*) Initial propeller of infinite diameter, infinitely small projected-area ratio and zero tip speed, with which all of the work delivered to the screw is delivered by it as direct thrust. The real slip of this screw up to the yield point of the column of water upon which it operates is zero; (*b*) Ultimate screw of finite diameter but infinitely small pitch, projected-area ratio of unity. In this latter case, the propeller has become a disc, all work delivered to the screw is consumed by it in rotation. The propulsive efficiency is zero and the slip is zero. The indicated thrust per square inch of disc area is equal to the indicated thrust per square inch of projected area.

On the curves of *P. T.* for .32 and .624 plot the points corresponding to an *I. T.* per square inch of projected area of 1, 2, 3, etc., pounds and through points of equal *I. T.* per square inch of projected area, draw curves perpendicular to the axis of *Y* at the intersection with *Y* and cutting the axis of *X* at an *I. T.* per square inch of disc area equal to the *I. T.* per square inch of projected area.

If the values $\frac{P.T.}{I.T.}$ as indicated for the primary propeller corresponding to the axis of *Y*, are laid down with *P. T.* per square inch of projected area as abscissæ and percentages as ordinates, it will be found that the percentage value, the propulsive coefficient, falls rapidly to quite a low value of *P. T.*, approximately 3.7 pounds per square inch of projected area, when it holds a nearly constant value until an abscissæ value of approximately 7.4 pounds has been reached, when the rate of decrease of the propulsive coefficient becomes accelerated.

Should a curve of apparent slips be plotted, it would be seen that the slip rises very rapidly at first, then for a period the rate of increase of slip becomes very considerably decreased

until a *critical point* is reached when the rate of slip increase becomes accelerated and continues increasing until cavitation sets in, when the rate of increase becomes very large. This *critical point* will be found to coincide with a line parallel to axis of X, its intersection A with the axis of the ordinates corresponding with the point A of the enveloping curve of propulsive coefficients shown on Plate 3, and to a $P.\ T.$ of 3.7 pounds to 7.4 pounds.

After passing the $P.\ T.$ corresponding to this line the columns of water under the projected area of each blade give way more and more as the propulsive thrust increases, until it finally begins to break, the head not being sufficient to provide the necessary velocity of flow, and cavitation begins.

Following will be found a table which may serve as a guide to judge where we must stop in $P.\ T.$ to guard against heavy vibrations of the screws occurring, due to the rupturing of the water columns. This table only applies to the lower values of projected-area ratios.

Vessel.	$\frac{I.\ T.\text{ per sq. in.}}{D.\ A.}$	$\frac{I.\ T.\text{ per sq. in.}}{P.\ A.}$	$\frac{P.\ A.}{D.\ A.}$	$\frac{P.\ T.\text{ per sq. in.}}{P.\ A.}$	$\frac{E.\ T.\text{ per sq. in.}}{P.\ A.}$	Vibration.
Delaware	4.64	14.15	.328	9.17	10.58	Light
West Virginia	4.35	15.4	.282	9.52	11.89	Light
Maryland	4.57	16.27	.281	9.81	12.43	Moderate
California	4.86	15.78	.308	8.68	10.33	Heavy
Virginia	4.6	15.17	.303	9.10	11.15	Considerable
Michigan	4.1	13.1	.315	9.10 or better	10.36	Light
Salem	9.0	16.56	.52	9.87	12.26	Moderate

$E.\ T. = \frac{P.\ T.}{1 - s}$ where s = apparent slip expressed as a decimal.

A propulsive thrust ($P.\ T.$) of 8.7 pounds per square inch of projected area of $P.\ A.$ ratio of .32 has been taken as a safe upper limit of design for that ratio of projected to disc area, beyond which we may not go without expecting vibration to occur, very slight at first and increasing in intensity as the $P.\ T.$ increases.

The commonly accepted formula for effective thrust, which, in the foregoing table, is in the column marked *E. T.*, is

$$E.T. = \frac{E.\ H.\ P.\ (\text{all appendages}) \times 33{,}000}{\text{Speed in knots per hr.} \times 101.33 \times \text{Proj. Area in sq. in.}}$$

$$= \frac{E.H.P. \times 33{,}000}{101.33 \times S. \times P.A.}$$

$$= \frac{33{,}000 \times I.H.P. \times \text{Propulsive Coefficient}}{S \times 101.33 \times P.A.}$$

$$= \frac{33{,}000 \times I.H.P. \times P.C.}{S \times 101.33 \times P.A.}$$

The term propulsive thrusts used are designated by the following:

$$P.T. = \frac{E.H.P. \times 33{,}000}{P \times R \times P.A.} = \frac{I.H.P. \times 33{,}000 \times P.C.}{P \times R \times P.A.},$$

but $101.33 \times S = P \times R \times (1 - s)$ where s is the apparent slip, then

$$E.T. = \frac{E.H.P. \times 33{,}000}{P \times R \times P.A. \times (1 - s)} = \frac{I.H.P. \times 33{,}000 \times P.C.}{P \times R \times P.A. \times (1 - s)}$$

$$= \frac{P.T.}{(1 - s)}$$

As the apparent slip at which proper propulsive efficiency can be obtained, with a propeller working in the wake of a vessel, depends upon the fineness of the lines of that hull, becoming higher as the lines become finer, it is readily seen how impossible it is to give a fixed value to the limiting value of *E. T.* as this value changes with variation in fineness of hull and with the projected-area ratio of the screw itself.

Plate 3. Chart of Propulsive Coefficients.—This chart is derived directly from the propulsive thrusts by dividing the ordinate values of propulsive thrust per square inch of *P. A.* by the indicated thrust per square inch of *P. A.* corresponding to this propulsive thrust for each value of *P. A.* ÷ *D. A.*, and plotting the results with per cent. propulsive coefficients

as ordinates and pounds indicated thrusts per square inch of disc area as abscissæ.

It will be noted how all the resultant curves of propulsive coefficients become tangent to an enveloping curve at some part of their course. Also how this enveloping curve cuts the axis of Y at the point A where the propulsive efficiency is approximately 73.2 per cent. The point A corresponds to the critical point A referred to previously. This enveloping curve may be called the locus of propulsive coefficients for the critical thrusts.

As the narrowest practical blade which can be built is probably not much less than .20 projected-area ratio, it is seen that the maximum practical propulsive coefficient at the point of design, the enveloping curve at this ratio, cannot exceed about 71½ to 72 per cent.

In the work of design the enveloping curve $A-E$ may be, but the modified curve A–D, shown on Plate 5, usually is used. In cases where low tip speeds are required the resultant primary propellers obtained will have a projected-area ratio of .2, but in such cases the diameters obtained are usually greater than can be swung under the circumstances, and the final resultant propeller will have less diameter and much greater projected-area ratio than, although having the same total projected area as, the theoretically desirable primary propeller obtained from the charts.

It has already been pointed out that as the propulsive coefficients used for the basic propeller of large projected-area ratio, and as this propeller was the average wing propeller of a three-shaft ship in which the center screw was of the same dimensions as the wing screws, and, therefore, not the best for the conditions under which it operated, the propulsive coefficient curve for this basic screw is probably a little low. This small error grows less as the projected-area ratios decrease towards the .32 ratio, where we may consider that no error exists. For ratios less than .32, the chart will then give very slightly higher efficiencies than can be realized, this error increasing the further we decrease the ratio below .32.

It should be further pointed out that where we have a vessel with three or four shafts, and with engines so arranged that there is a great variation in the distribution of power on the shafts due to various changes made in the number of turbines being used, with the different possible combinations provided for economical cruising, a departure from the propulsive coefficient curves at the low values of thrust may be looked for. This is especially the case where a four-shaft ship is in question. As the error of the chart is produced by using the propulsive coefficients of a three-shaft vessel, such vessels should fit it closely.

The effect of this variable distribution of power on the different shafts upon the propulsive coefficients will be shown later.

Plate 4. Chart of Tip Speeds.—In the original chart of tip speeds, which was developed from the tip speeds for propellers of .32 and .54 projected-area ratios, it was assumed that for equal indicated thrusts per square inch of projected area, the tip speeds would increase in proportion to the projected-area ratios in passing through the points of equal indicated thrusts per square inch of projected area on the tip-speed curves of .32 and .54 projected-area ratios.

Later data for higher projected-area ratios than .54 show this not to be the case, but rather that the locus curves of tip speeds for equal indicated thrusts per square inch of projected areas are curves springing from the origin, tip-speed zero, rising rapidly to an enveloping curve of tip speeds to which each curve becomes tangent, and then falling off to the right of this curve.

In developing these curves of tip speeds, the tip speeds for projected-area ratios of .32, .52 and .624 were used.

Plates 5, 6 and 7. Design Charts of Thrusts and of Apparent Slips.—The method of obtaining the slips for different blocks has been thoroughly described in previous articles and will not be repeated here.

The method of laying down the slip curves for projected-area ratios other than for the .32 and .624 ratios differed,

however, from the method used formerly, in that instead of drawing a straight line through and connecting points of equal indicated thrust per square inch of projected area on each basic slip curve and interpolating on this line for the apparent slips of other ratios, this line is replaced by a curve starting from the origin and passing through the points of equal indicated thrust per square inch of projected area and returning to the axis of abscissæ where the propeller has become a solid disc of no pitch and zero slip, and the indicated thrusts per square inch of projected and of disc areas are equal.

On these curves of equal thrusts the interpolations for the other projected-area ratios are made.

Having drawn in these various slip curves, the locus curves *A–D* corresponding to the same lettered curve of the propulsive coefficient on Plate 5 are plotted in.

It must not be understood that the curve *A–D* must not be exceeded. This is not the case, but for lack of data above this curve *A–D* at the higher projected-area ratios, a satisfactory limit could not be laid down. The enveloping curve *A–E* agrees nearly with *A–D* up to about a projected-area ratio of .50, above that practice follows *A–D* which is lower than *A–E*.

The charts, Plates 5, 6 and 7, derived from the slip curves, also include curves of tip speeds corresponding to the curve of design *A–D*. They can be used alone for designing the screw after checking the block coefficient by Chart 1, and the appendage resistance by Chart 2. Thus, for Plate 5,

$$\text{Propulsive Coefficient} = \frac{\text{Point on Curve 3}}{\text{Corresponding Point on Curve 1}}.$$

$$\text{Apparent slip expressed as a decimal} = 1 - \frac{\text{Point on Curve 3}}{\text{Corresponding Point on Curve 2}}.$$

Curve 1 = pounds indicated thrust per square inch of *P. A.* $= I.\ T._p$.

Curve 2 = pounds effective thrust (*E. T.*) per square inch of projected area = *P. T.* ÷ (1 − slip expressed as a decimal).

Curve 3 = pounds propulsive thrust (*P. T.*) per square inch of *P. A.* = *I. T.*$_p$ × propulsive coefficient.

Curve 4 = Curves of *I. T.* per square inch of disc area (*I.T.*$_D$).

Curve 5 = Curves of Tip Speeds for *A–D*.

Curve 6 = Curves of Prop. Coef. for *A–D*.

These curves are all erected on projected-area ratios as abscissæ.

Curves 2 are curves of *E. T.* for different block coefficients varying by .05 from .2 to .9.

The full line curves are those which are usually followed in practice. The extended portions shown in dotted lines are the approximate thrusts corresponding to the enveloping curve, *A–E*, of propulsive coefficients.

By examining Plate 3, it will be noted that the propulsive coefficient curves for all projected-area ratios reach their maximum before becoming tangent to *A–E*. After touching this curve they fall away rapidly. The writer is of the opinion that this curve of the envelope marks the region in which cavitation begins. At first so slight is it that it is not noticeable. It gradually increases until, after slightly exceeding *A–D* with the lower projected-area ratios, the vibrations due to it have become so well defined as to be marked light to moderate. The line of full cavitation is somewhere beyond the limit *A–E*. If this is true, then in designing we are always absolutely safe in designing on the curves *A–E*, as there is no reason to suppose that the limit *A–E* cannot as safely be exceeded at the higher values of projected-area ratio as at the lower, where we may go as high as *A–D* without vibration setting in. Again, if this assumption of *A–E* is a safe one, it is seen at once from the curves 2 that for a very fine block coefficient, .2, and a high projected-area ratio, .65, the *E. T.* may equal, at the lowest, 15.3 pounds without cavitating.

From an examination of the curves 2, it will be seen that

as the block of the ship fines and, as the projected-area ratio of the propeller increases, the *E. T.* increases. We find it very low for a projected-area ratio of .15 and a block coefficient of .9, as low as 6.4 pounds in fact, while, as pointed out in the preceding paragraph, for a block coefficient of .2 and a projected-area ratio of .65 it has risen to at least 15.3. The qualifying words "at least" are used for the reason that the dotted lines are drawn tangent to the points of inflexion of the curves 2, where they swing over to follow the line of general practice, *A–D*.

TABLE OF ACTUAL *E. T.* REALIZED IN SERVICE FOR VARIOUS BLOCKS AND HIGH PROJECTED-AREA RATIOS—PROPELLERS RUNNING SMOOTHLY.

Screw.	$\frac{P.A.}{D.A.}$	Nominal Block.	$\frac{B}{L.W.L.}$	Block by Plate 1.	Actual *E. T.*	*E. T.* By Curve *A–D*.	*E. T.* By Curve *A–E*.	Form of *P. A.*
No. 1..	.52	.407	.117	.415	12.35	10.9	10.9	Standard
No. 2..	.538	.414	.0899	.35	13.48	11.55	11.55	Narrow at tip
No. 3..	.542	.414	.0899	.35	13.45	11.65	11.65	Narrow at tip
No. 4..	.564	.414	.0899	.35	12.95	12.16	12.30	Narrow at tip
No. 5..	.584	.414	.0899	.35	12.75	12.50	12.95	Standard
No. 6..	.624	.403	.0900	.35	12.3	13.00	14.2	Standard
No. 7..	.60	.410	.0899	.35	13.17	12.70	13.4	Standard

It should be borne in mind that while we may in the values of *E. T.* considerably exceed the values of *E. T.* corresponding to *A–E* and *A–D*, as shown by experience for the lower projected-area ratio, vibration of considerable magnitude may be met with at high thrusts with the higher projected-area ratios, even though *A–E* may not be exceeded. These vibrations may, however, be due to an entirely different cause than cavitation. They may be, and probably are, caused by the tips of the blades passing the hull in too close proximity to it, and at a very high speed. There is no doubt in the mind of the writer that many cases of heavy vibration at high speeds are due to this cause.

An explanation offered for the effect of increased projected-area ratio in preventing cavitation is that, in passing through the water, the water cascades around the leading edge of the

blade thus causing an amount of the leading portion of the blade face to become ineffective as thrust area.

In order to counteract this loss of effective driving face of the blade it then becomes necessary to widen the blade, thus increasing the projected-area ratio. This explanation is hardly satisfactory, for if the model propeller has no speed of rotation but be moved through the water by the carrier at a good rate of speed, this same cascading of the water around the leading edge will undoubtedly occur, and will continue until the product of pitch × revolutions of the propeller equals the speed of advance.

It will be shown, later, how the equation for the diameter of the propeller takes the following form:

$$D = \sqrt{\frac{2.88\ E.\ H.\ P.}{\frac{P.\ A.}{D.\ A.} \times S \times E.\ T.}}, \text{ where}$$

D = diameter of the propeller in feet.

$E.\ H.\ P.$ = effective horsepower on one screw for speed S.

S = speed in knots per hour.

$\frac{P.\ A.}{D.\ A.}$ = Projected-area ratio.

$E.\ T.$ = Effective thrust in pounds per square inch projected area.

From this formula we see that for equal values of $E.\ T.$ and of speed the projected-area ratio of the screw varies inversely as the square of the diameter; in other words, in a series of propellers designed to deliver the same $E.\ H.\ P.$, under similar conditions of speed and $E.\ T.$ per square inch of projected area, the projected-area ratios of the screws as compared with each other will vary inversely as the squares of the diameters of the screws.

Plate 8. Curves showing Developed-Area Ratios corresponding to different Projected-Area Ratios of standard form blades for various values of Pitch divided by Diameter, also Curves of Standard Projected-Area Forms.—This chart has been redrawn and the projected-area ratio forms are given from .15

to .70 at intervals of increase of .05. The developed-area ratios corresponding were obtained by actual development of the blades on the draughting board and are as nearly correct as it is possible to make them.

In laying down a propeller from data obtained from the charts, the projected areas necessary will be that included between the tip and the standard hub given on Plate 8, any increase in area obtained by decreasing the diameter of the hub to a less percentage of the total diameter of the screw than that given by Plate 8 being neglected.

Below is given a table of multipliers for obtaining the chords of the arcs of the half projected-area widths at different radii for aid in laying down standard forms.

MULTIPLIERS FOR CHORDS OF ½ RADIAL ARCS OF BLADES OF STANDARD FORM OF PROJECTED AREA, THE RADIUS OF THE PROPELLER BEING UNITY.

R radius of Screw; M multiplier; Chord of ½ arc width of blade $= M \times R$.

P. A.	*M* at.											
D. A.	.2 *R*	.3 *R*	.4 *R*	.5 *R*	.6 *R*	.7 *R*	.8 *R*	.9 *R*	.925 *R*	.95 *R*	.975 *R*	.9875 *R*
.15	.04	.063	.085	.104	.118	.127	.125	.104	.093	.078	.058	.045
.20	.053	.074	.114	.128	.158	.169	.168	.138	.125	.106	.078	.058
.25	.068	.105	.142	.172	.198	.212	.210	.175	.156	.134	.097	.071
.30	.081	.128	.170	.208	.235	.252	.257	.209	.190	.161	.117	.088
.35	.095	.148	.199	.241	.275	.295	.293	.245	.222	.190	.138	.100
.40	.108	.168	.226	.275	.314	.336	.333	.278	.250	.214	.158	.116
.45	.122	.190	.253	.308	.352	.378	.376	.313	.272	.243	.177	.126
.50	.135	.210	.280	.340	.389	.418	.417	.346	.310	.269	.195	.140
.55	.147	.230	.306	.372	.427	.460	.458	.385	.345	.296	.218	.160
.60	.158	.249	.332	.405	.464	.500	.498	.418	.377	.322	.240	.178
.65	.171	.270	.359	.435	.499	.540	.539	.448	.403	.345	.258	.198
.70	.183	.287	.383	.465	.533	.578	.577	.482	.437	.375	.280	.226

It will be noticed that with all of these forms the greatest width in each case is at the same distance from the axis of the hub, approximately .7 the radius of the propeller from the axis.

The necessity of adhering to standard projected-area forms in order to obtain graphical or other methods of design will be easily understood by examining the following figures, 1, 2, and 3, showing forms of projected and developed areas for the Standard, the Barnaby and the Taylor forms of blades.

Fig. 1 shows the standard form of projected area marked a, and the developed areas for pitch ratios of .8 and of 2, marked a' and a'' respectively, together with the corresponding projections on the fore-and-aft plane.

Figs. 2 and 3 show equal developed areas with Fig. 1, and the corresponding projected areas on the disc and on the fore-and-aft plane, for Barnaby's and for Taylor's blades respectively.

The dotted forms shown on Figs. 2 and 3 are the projected-area forms of Fig. 1.

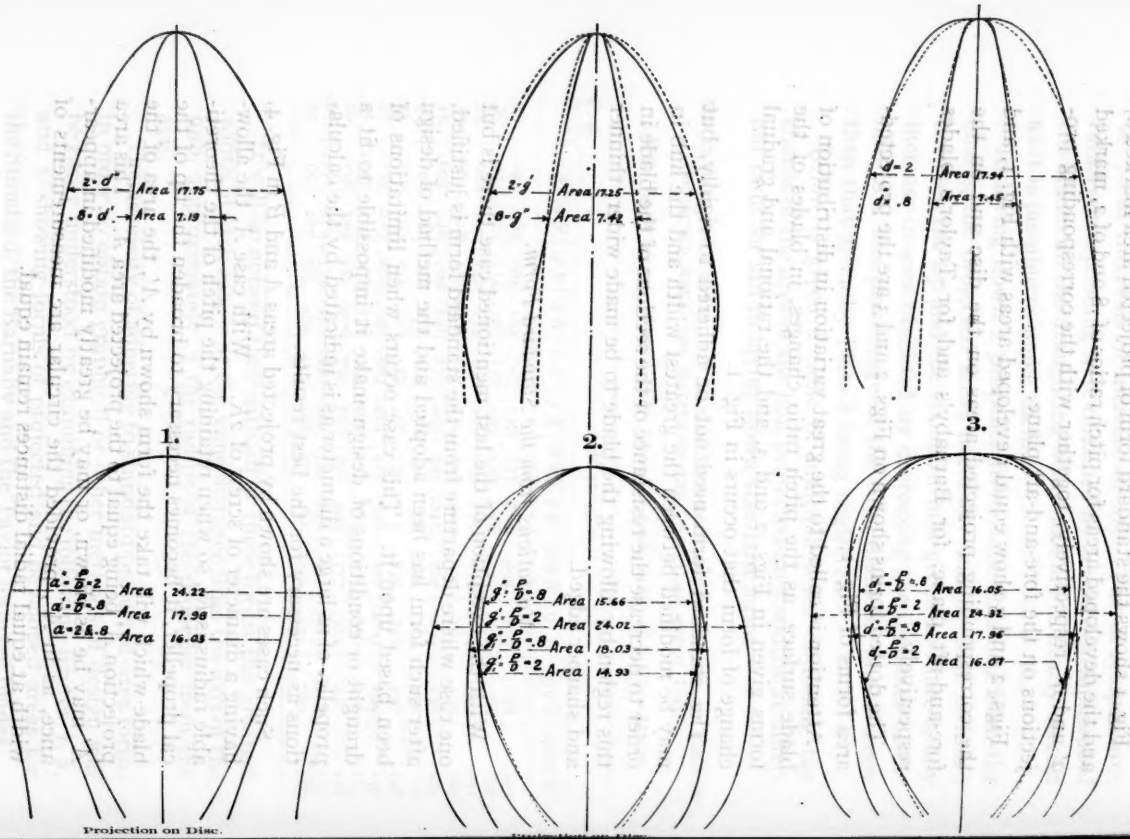
Attention is called to the great variation in distribution of blade surface, as the pitch ratio changes, in blades of the forms given in Figs. 2 and 3, and the rational and gradual change of form that occurs in Fig. 1.

The standard form need not be adhered to rigidly, but may be modified between the greatest width and the hub in order to decrease the resistance of the sections of the blade in this region by allowing the blade to be made wider, thinner and sharper edged.

Variations from the Standard Form.

With the exception of the last mentioned case there is but one case where departure from the standard form is justified, after such form has been adopted and the method of design been based upon it. This case occurs when limitations of draught or conditions of design make it impossible to fit a propeller of as large a diameter as is indicated by the calculations as necessary for the best results.

Such cases are shown by projected areas A and B in Fig. 4, having a diameter of screw of $2R$. With case A, the allowable radius is R', so when retaining the pitch of the theoretical propeller, it becomes necessary to broaden the tip of the blade which will take the form shown by A', the area of the projection A' being equal to the projected area A. This area A' may be as shown, or may be greatly modified in appearance, as in A'', provided the circular arc measurements of width at equal radial distances remain equal.


2·d" Area 17.75
.8·d' Area 7.19
1.
a'·P/D=2 Area 24.22
a'·P/D=.8 Area 17.98
a=2&.8 Area 16.03
Projection on Disc.
2·g' Area 17.25
.8·g" Area 7.42
2.
g'·P/D=.8 Area 15.66
g'·P/D=2 Area 24.02
g'·P/D=.8 Area 18.03
g'·P/D=2 Area 14.93
d=2 Area 17.54
d'=.8 Area 7.45
3.
d'·P/D=.8 Area 16.05
d=P/D=2 Area 24.32
d'·P/D=.8 Area 17.96
d·P/D=2 Area 16.07

Where the difference between the calculated and the allowed diameters is large, the resultant blade would have an abnormal form, as in B'. This form is often met with in motor-boat propellers, disguised as B''. Patents have been allowed on this form, and great claims are made for it on the grounds of high efficiency, when in reality its greater effi-

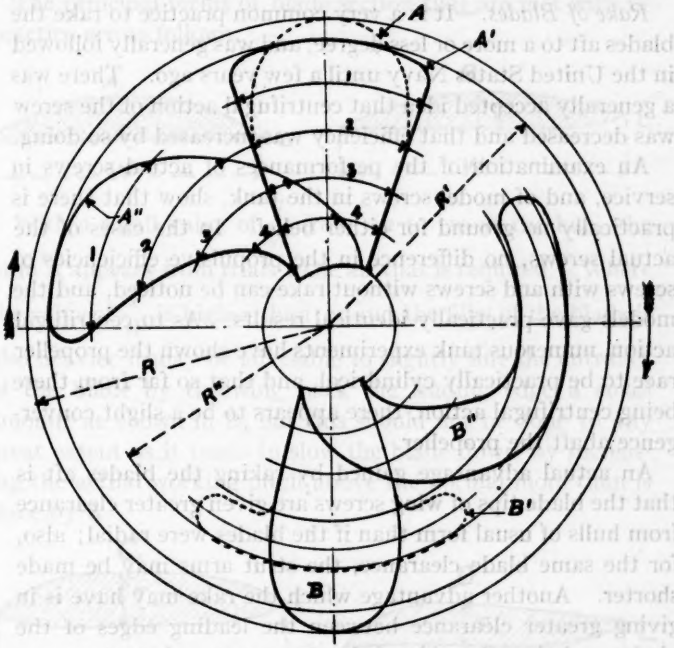


Fig 4.

ciency over a blade of ordinary form is caused by its approximation in amount and distribution of area to those of a propeller having the proper dimensions for the work which it is called on to do. In place of this tip broadening, however, the problem may be solved for "reduced diameter," as in Problem 1, and the standard form of projected area be adhered to.

Blades thrown to the side, as in A'', are used to reduce

vibration in cases where a rapid-running screw operates close to the strut or stern post. As a general rule, though, they are undesirable, as the form is weak and the blade must be made extra heavy at the root in order to provide the necessary rigidity to insure against change of pitch, due to springing of the blade when subjected to heavy thrusts.

Rake of Blades.—It is a very common practice to rake the blades aft to a more or less degree, and was generally followed in the United States Navy until a few years ago. There was a generally accepted idea that centrifugal action of the screw was decreased and that efficiency was increased by so doing.

An examination of the performances of actual screws in service, and of model screws in the tank, show that there is practically no ground for either belief. In the cases of the actual screws, no difference in the propulsive efficiencies of screws with and screws without rake can be noticed, and the models gave practically identical results. As to centrifugal action, numerous tank experiments have shown the propeller race to be practically cylindrical, and that so far from there being centrifugal action, there appears to be a slight convergence abaft the propeller.

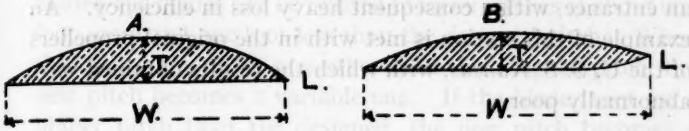
An actual advantage gained by raking the blades aft is that the blade tips of wing screws are given greater clearance from hulls of usual form than if the blades were radial; also, for the same blade clearance, the strut arms may be made shorter. Another advantage which the rake may have is in giving greater clearance between the leading edges of the blades and the after side of the stern post and struts.

Radial blades, in addition to being as efficient as those with rake, are more easily machined, have less total developed area for equal projected area, and therefore less surface friction, are stiffer and lighter; also, the stresses in the blades due to centrifugal action are less. With propellers of high speed of revolutions, this latter point is very important, and for such screws the blades should never rake.

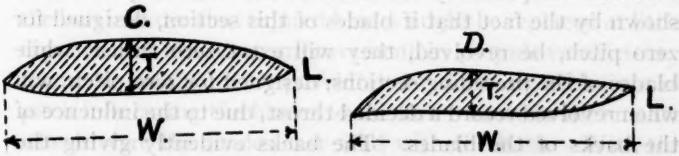
Blade Section.—Care must be exercised in choosing the shape of blade section in order not to cause heavy losses by

interfering with the ease of flow of water past the blade, thus causing eddies and the formation of cavities by too abrupt a change in the direction of flow of the water; we should also avoid, as much as possible, the use of sections of such form as would make the actual working pitch of the blade widely divergent from the nominal designed pitch.

The principal forms of blade section that are met with in practice are as follows:



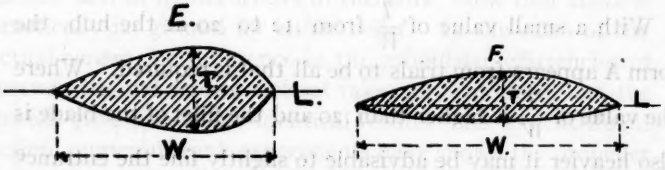
With a small value of $\frac{T}{W}$ from .12 to .20 at the hub, the form A appears from trials to be all that is required. Where the value of $\frac{T}{W}$ is higher than .20 and the fillet of the blade is also heavier it may be advisable to slightly fine the entrance of the blade by throwing back the leading edge a small amount as shown in B, but this should not be done to any great extent as it tends to slow the blade down by increasing the actual working pitch above the normal more than is done by A.



Section C with the following edge of the blade thrown back, the leading edge being either similar to A or thrown back as shown, is considered by the writer to be a decided mistake, since, as the water travels along the driving face of the blade from the entering to the leaving edge, there may exist a tendency for it to break contact with the blade face. It was to guard against this tendency of the water to leave the blade face that axially-expanding pitches of blades were used.

If the following edge of the blade is thrown back as in C, the face of the blade is deliberately drawn away from the water and a cavity at this edge will result with consequent eddying effect and resultant vibration and loss in efficiency.

In section D, the leading half of the back of the blade has a pitch such that its slip equals the real slip of the screw. This form is theoretically correct, but in practice, unless the blades are very wide, it gives too thick a blade and too blunt an entrance, with a consequent heavy loss in efficiency. An example of this section is met with in the original propellers of the U. S. S. *Kansas*, with which the results obtained were abnormally poor.



Blades of section E, with the metal of the blade divided evenly on each side of the nominal pitch surface or plane, appear to offer less resistance to turning than any of the other sections, due probably to the fact that the real pitch of the blade is also probably the same as the nominal pitch, as is shown by the fact that if blades of this section, designed for zero pitch, be revolved, they will exert zero thrust, while blades of the preceding sections, designed for zero pitch, will when revolved record a decided thrust, due to the influence of the backs of the blades. The backs evidently giving the blade a working pitch greater than the nominal pitch cause them to exert a thrust. While it requires less power to turn blades of section E, the resultant thrust per revolution is much lower and the apparent slip is much higher than with blades of the same nominal pitch but of different section.

With blades having sections similar to F, the same results are obtained as with section E, but in a less degree.

Effect of Changing the Pitch of Blades from the Designed Pitch by Twisting the Blades on the Hub.

Very often upon the trial of a vessel, results indicate that improvement is possible if the propeller blades be set to a higher or lower pitch than that of the designed driving face. In order to provide for such a change the bolt holes in the blade pads are made oval, so that ordinarily with large blades, the blades can be twisted to mean pitches of about one foot more and one foot less than the designed pitch. When the blades are set for any other than the designed pitch, the new pitch becomes a variable one. If the blade is set for a higher pitch than the designed, the new pitch becomes a radially expanding one, increasing from the hub towards the tips, while if the new pitch is lower than the designed, the pitch will decrease radially from the hub to the tips.

The change caused by alterations in pitch may be obtained from the following table (see Peabody's Naval Architecture) by multiplying the original pitch ÷ diameter by the factors given in the table for the small angle through which the blade is twisted. (Table, page 596.)

The Hub.—In designing screw propellers it was, up to the advent of the turbine, the custom to almost invariably design propellers of large diameter with the blades detachable from the hub in order that injured blades might be replaced at little expense and also that improvement in propulsive efficiency might be sought for by providing for slight modifications of pitch in securing the blades to the hub. Only in the smaller wheels were the blades cast solid with the hubs. With the pitch ratios ordinarily in use with the comparatively high pitch, slow-turning reciprocating-engine propellers, where the hub diameters varied from 20 per cent. to 27.8 per cent. of the diameter of the screw in built-up wheels, the pitch angles at the hub ranged from about 50 degrees to 58 degrees, while with the solid propellers with hubs varying in diameter from $14\frac{3}{4}$ per cent. to $18\frac{1}{4}$ per cent. of the diameter of the propeller, the pitch angle varied from 67 degrees to 76 degrees at the hub.

Angle the Blade is Twisted.		1°.		2°.		3°.		4°.		5°.		6°.	
Pitch Ratio.	Diameter Ratio.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.
0.8	1.25	1.07	0.93	1.15	0.85	1.22	0.78	1.29	0.71	1.37	0.64	1.45	0.57
0.9	1.11	1.07	0.94	1.14	0.87	1.20	0.81	1.27	0.75	1.34	0.68	1.41	0.61
1.0	1.00	1.06	0.94	1.12	0.88	1.19	0.82	1.25	0.76	1.31	0.70	1.38	0.65
1.1	0.90	1.05	0.94	1.11	0.89	1.17	0.84	1.23	0.78	1.29	0.73	1.35	0.68
1.2	0.83	1.05	0.94	1.10	0.89	1.16	0.84	1.21	0.79	1.27	0.74	1.32	0.70
1.3	0.77	1.05	0.95	1.10	0.90	1.15	0.86	1.21	0.81	1.26	0.76	1.31	0.72
1.4	0.71	1.05	0.95	1.09	0.91	1.14	0.86	1.20	0.82	1.24	0.77	1.30	0.73
1.5	0.66	1.05	0.96	1.09	0.91	1.14	0.87	1.19	0.83	1.24	0.78	1.29	0.74
1.6	0.62	1.04	0.96	1.09	0.91	1.13	0.87	1.18	0.83	1.22	0.79	1.28	0.75
1.7	0.59	1.04	0.96	1.08	0.92	1.13	0.88	1.17	0.84	1.22	0.80	1.27	0.76
1.8	0.55	1.04	0.96	1.08	0.92	1.13	0.88	1.17	0.84	1.21	0.81	1.26	0.77
1.9	0.52	1.04	0.96	1.08	0.92	1.12	0.88	1.16	0.85	1.20	0.81	1.25	0.78
2.0	0.50	1.04	0.96	1.08	0.92	1.12	0.88	1.16	0.85	1.20	0.81	1.25	0.78
2.1	0.47	1.04	0.96	1.08	0.92	1.12	0.88	1.16	0.85	1.20	0.82	1.24	0.79
2.2	0.45	1.04	0.96	1.07	0.93	1.11	0.89	1.15	0.85	1.20	0.82	1.24	0.79
2.3	0.43	1.03	0.96	1.07	0.93	1.11	0.89	1.15	0.86	1.19	0.83	1.24	0.79
2.4	0.42	1.03	0.96	1.07	0.93	1.11	0.89	1.15	0.86	1.19	0.83	1.24	0.80
2.5	0.40	1.03	0.96	1.07	0.93	1.11	0.90	1.15	0.86	1.19	0.83	1.23	0.80

Desirable as it is to reduce the diameter of the hub to that of the strut boss in order to avoid eddying between the boss and strut, it is not always possible to do this with the hubs of built-up propellers. The seating of the propeller-blade pad in the hub must be circular to permit of pitch adjustments, the hub must be spherical to maintain its symmetry of outline when variations of pitch are made, and the seating must be of sufficient diameter to accommodate a proper number of holding bolts of sufficient strength; finally, the blade pad must be of sufficient width to accommodate a blade having such a ratio of thickness to width as will prevent excessive blade resistance.

The effect of the above requirements when met for tubine-

driven propellers of large diameter and low-pitch ratio was to throw the effective blade areas too far out from the axis of the hub, thus leading to serious increase in blade friction at the high tip speeds employed, and also to bring the 45-degree pitch angle of the helical surface well within the surface of the hub. In the few cases coming to the notice of the writer in which detachable blades were used for these high-speed turbine screws, the results obtained were poor, but there were other conditions existing in these cases which may have been responsible for the poor propulsive efficiency realized.

Location of Blade on Blade Pad.

In order to provide sufficient space for the blade bolts to pass through the pad without cutting into the true working face or the working section of the blade, disregarding the fillet, it is very often necessary to shift the blade on the pad so that the blade axis does not coincide with the axis of the pad. To do this, the blade axis may be shifted forward or aft of the pad axis along the axis of the hub, or it may be swung around the hub to an angular position with the axis of the pad, or we may combine the two shifts.

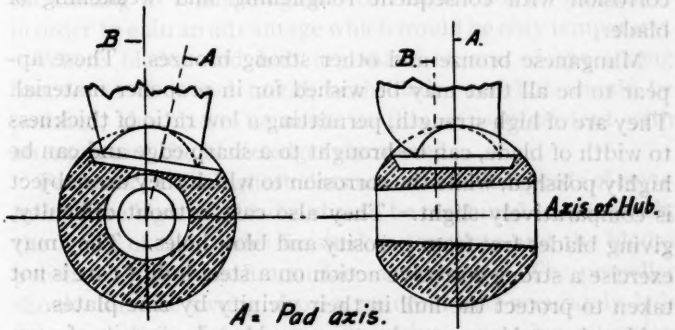


FIG. 5.

Whatever is done, in this way, care must be taken that the axis of the generatrix shall always remain coincident with the axis of the hub. The writer has met with many cases where, in order to give sufficient space for the blade bolts, the blade has been moved to one side or the other of the pad axis, the axis of the blade remaining parallel to the axis of the pad, but the axis of the generatrix becoming transferred several inches to one side of the hub axis to which it remains parallel.

Material of Blades.—The materials of which propeller blades are made are cast iron, cast steel, forged steel, manganese or some other strong bronze, and monel metal.

Cast iron is used for the blades of propellers which work under conditions rendering them very liable to strike against obstructions. When so striking, the cast iron being weak, the blade breaks and, by so breaking, saves the shafting or the engine. Its disadvantages are extreme corrosion in sea water, heavy blade sections and blunt edges due to the weakness of the metal.

Cast steel is stronger than cast iron, but has the same disadvantages although in a lesser degree.

Forged steel was formerly used in some instances for torpedo-boat propellers, but is not met with in present-day practice. It also possessed the disadvantage of excessive corrosion with consequent roughening and weakening of blade.

Manganese bronze and other strong bronzes. These appear to be all that may be wished for in propeller material. They are of high strength, permitting a low ratio of thickness to width of blade, can be brought to a sharp edge and can be highly polished, while the corrosion to which they are subject is comparatively slight. They also cast without difficulty, giving blades free from porosity and blow holes. They may exercise a strong corrosive action on a steel hull if care is not taken to protect the hull in their vicinity by zinc plates.

Monel metal is extremely strong and tough, permits of very light blade sections and sharp edges, takes a very high polish and is practically non-corrosive in sea water. These quali-

fications are all very desirable in a propeller metal. It has, however, the undesirable qualities of heavy and irregular warping of the blades when cooling in the mold, and a tendency to porosity around the tips and blade edges. On account of the tendency to warp, it is very difficult to insure the desired pitch unless the blade be cast with a large amount of waste metal which will permit of pitch correction in machining.

Material of the Hub.—Hubs are usually made of cast iron or semi-steel for cast-iron or cast-steel propellers; of semi-steel for the poorer classes of work, and of manganese bronze for the better class, with manganese bronze blades; of manganese bronze or of monel metal for monel-metal blades. Where the propeller is cast solid, of course the hub is of the same material as the blades.

General Requirements for Propellers.

For all propellers except those made of cast iron or cast steel, the blades should be polished in order to reduce surface friction. With cast-steel and cast-iron blades, however, as they are usually used for work where the speeds of revolution and tip speeds are low, the loss due to roughness of surface is not very high and it is preferable to retain the hard skin of the casting as a guard against corrosion than to sacrifice it in order to gain an advantage which would be only temporary.

Where blades and hub are made of one of the strong bronzes or of monel metal, both blades and hub should be polished, the blades be made as thin as is consistent with strength and the blade edges sharpened.

For work of the highest class and where the speeds of revolutions are high, the blades should be machined to true pitch, the backs of the blades finished to template and the blades polished to as smooth a surface as possible. The propeller should then be swung upon a mandrel and accurately balanced, as lack of balance will produce excessive vibration when the speeds of revolution are high. In some cases, in order to insure a smooth blade surface, bronze blades are

silver plated. This, however, seems of doubtful expediency. It insures smooth blades for the trial trip but it is doubtful if the silver plating would remain on the blades for any considerable length of time.

Number of Blades.—Froude in his experiments with model propellers in the model tank demonstrated that two-bladed propellers are slightly more efficient than three-bladed ones, and three-bladed than four-bladed ones. These results have since been verified by Taylor, who states that "efficiency was inversely as the number of blades."

Using three propellers, one of two blades, one of three, and one of four, with all blades of the same form and area, the total area of the four blades will be represented by unity, that of the three by .75, and of the two by .5. The resultant thrusts, however, will not be in these same proportions. That of the four-bladed will be unity, of the three-bladed .865, and of the two-bladed .65. The above results suppose the propellers to be alike in all respects but number of blades and total blade area.

A two-bladed propeller should never be used except for very small sizes, as it tends to produce heavy vibration. When used for large sizes it is usually in cases where the vessel is fitted with a lifting propeller which is drawn up into a propeller well when the vessel is under sail.

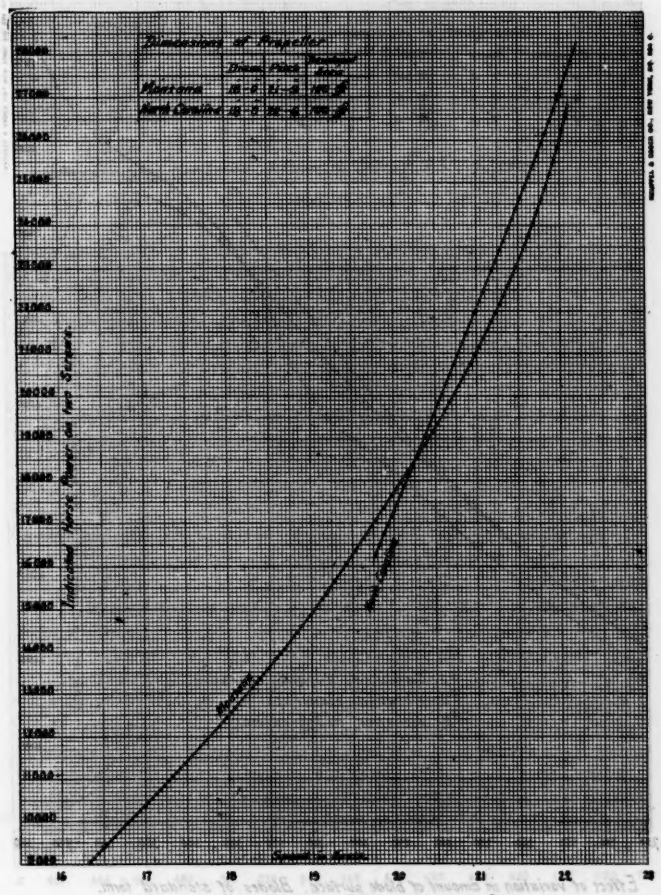
Three-bladed propellers appear to be the most desirable for general service. The blade area when distributed among three blades gives a wider and finer blade than if distributed among four, and where detachable blades are used the hub is smaller than with four blades.

Four-bladed propellers are recommended for use in cases where there is such a great difference between the maximum and minimum load conditions of the vessel that in the latter condition a greater portion of each blade is above the surface of the water during the upper half of its revolution. The greater number of blades in this case will tend to make the screw run smoother.

In some case of multiple screws, some of the screws have

been made three-bladed and the remainder four-bladed, in order to break up synchronism of vibration, and thus have the resultant vibration of the hull due to the propellers reduced to a minimum. The results obtained are stated to have been satisfactory.

PROPELLER DESIGN.

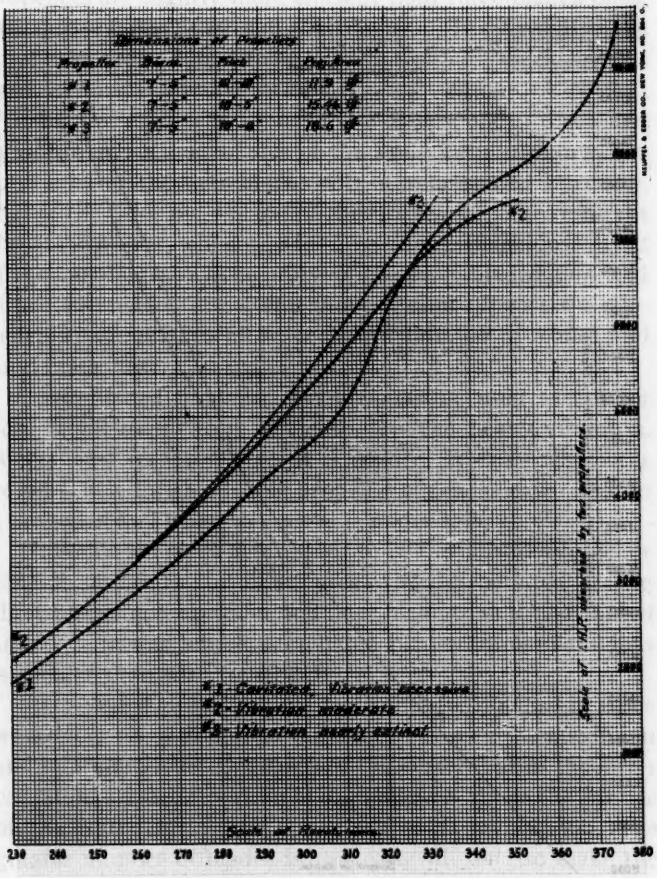


Montana, North Carolina, Effect of change in Pitch.

FIG. 6.

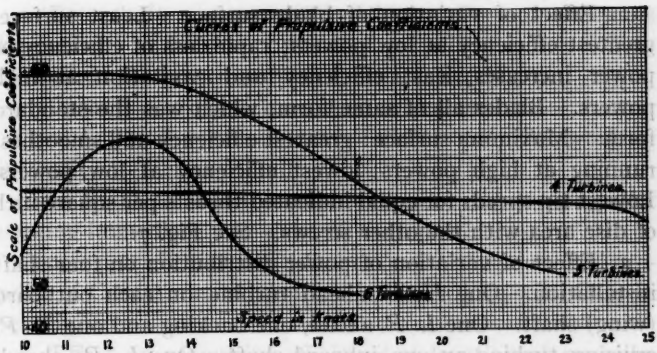
Some Points Governing Propulsive Efficiency.

1. Effect of excess pitch. Shown by Fig. 6.—Gain in propulsive coefficient at low powers. Loss in propulsive coefficient at high powers. Both sets of propellers having blades exactly alike.


#2
#1
#1. Contract. Vibration excessive.
#2. Vibration moderate.
#3. Vibration nearly optimal.
230 240 250 260 270 280 290 300 310 320 330 340 350 360 370 380

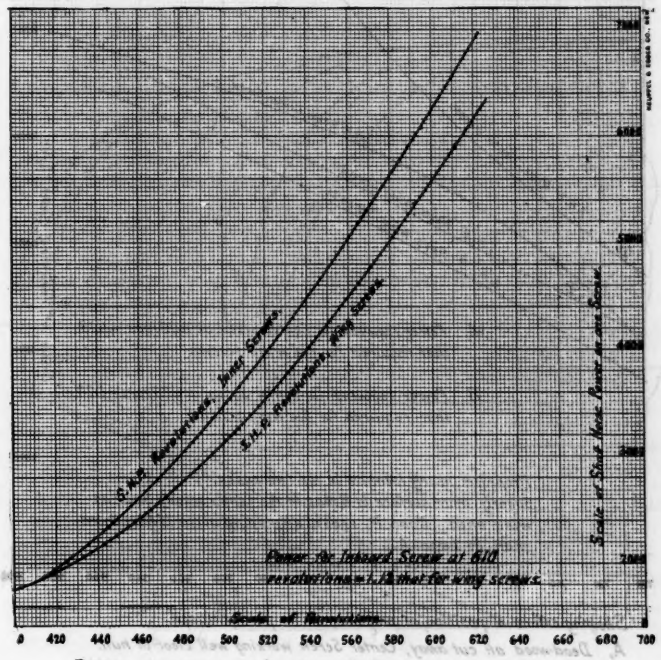
Effect of variation in amount of blade surface. Blades of standard form.

FIG. 7.


Curves of Propulsive Coefficients.
Scale of Propulsive Coefficients
4 Turbines
5 Turbines
6 Turbines
Speed in Knots
10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25

Effect of varying distribution of Power on Shafts. U.S.S. "Chester."

FIG. 8.

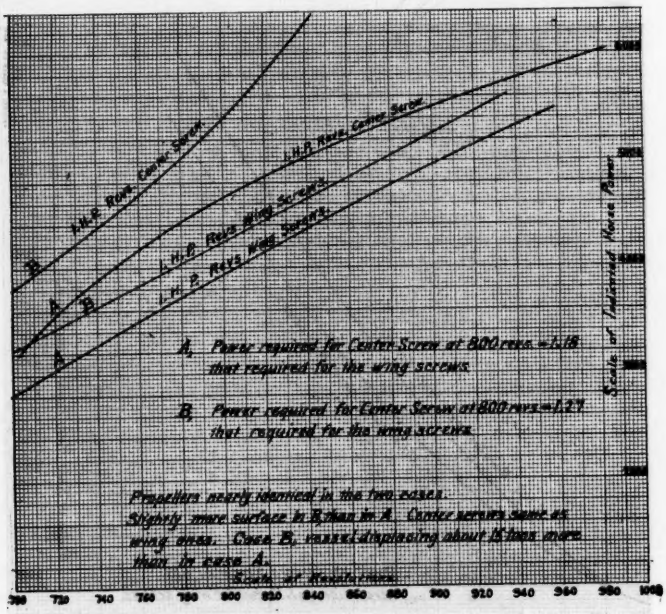

Scale of Shaft Horse Power as ran Scale
Power for Inboard Screw at 610 revolutions = 1.13 that for wing screws
Scale of Revolutions
0 420 440 460 480 500 520 540 560 580 600 620 640 660 680 700

Effect of position of propeller. 4 Shaft arrangement.

FIG. 9.

2. Effect of variation of blade surface.—Least surface: greatest efficiency at low powers; rapid loss of efficiency as power increases; least efficiency and cavitation at high powers. Blades all of same form, which was the standard form. Maximum surface: greatest efficiency and smoothest running at high powers; lowest efficiency at low powers. Lowest tip speeds for equal indicated thrusts per square inch of disc area with the other screws. See Fig. 7.

3. Effect of variation of power distribution on four-shaft installation.—One *H. P.* ahead turbine on each outboard (wing) shaft. One *L. P.* ahead, one backing and one *M. P.* cruising turbine on one inboard shaft. One *L. P.* ahead, one backing and one *H. P.* cruising turbine on other inboard shaft. See Fig. 8.


I.H.P. Revs. Center Screw
I.H.P. Revs. Wing Screws
I.H.P. Revs. Wing Screws
A, Power required for Center Screw at 800 revs. = 1.18 that required for the wing screws
B, Power required for Center Screw at 800 revs. = 1.27 that required for the wing screws
Propellers nearly identical in the two cases.
Slightly more surface in B than in A. Center screws same as wing ones. Case B, naturally displacing about 18 tons more than in case A.
Scale of Indicated Horse Power
Scale of Revolutions

Showing effect of position of propeller on its performance.
A, Dead-wood aft cut away, Center Screw working well clear of hull.
B, Center Screw working in wake of and close to stern post.

FIG. 10.

4. Four-shaft arrangement.—All propellers of the same dimensions. Effect of position of propellers in relation to the hull on the power and revolutions. See Fig. 9.

5. Cases A and B. Three-shaft arrangements.—

Case A. Dead wood cut away. Center propeller working in locality well clear of hull.

Case B. Dead wood carried well aft. Center propeller working immediately in wake of stern post.

Center propellers of same dimensions as wing ones and propeller of Case A almost identical in dimensions with Case B. See Fig. 10.

6. Effect on propulsive efficiency of location of propeller when operating in the wake of a full after-body.

An interesting problem of the above conditions has recently arisen in the case of a self-propelled barge constructed for the

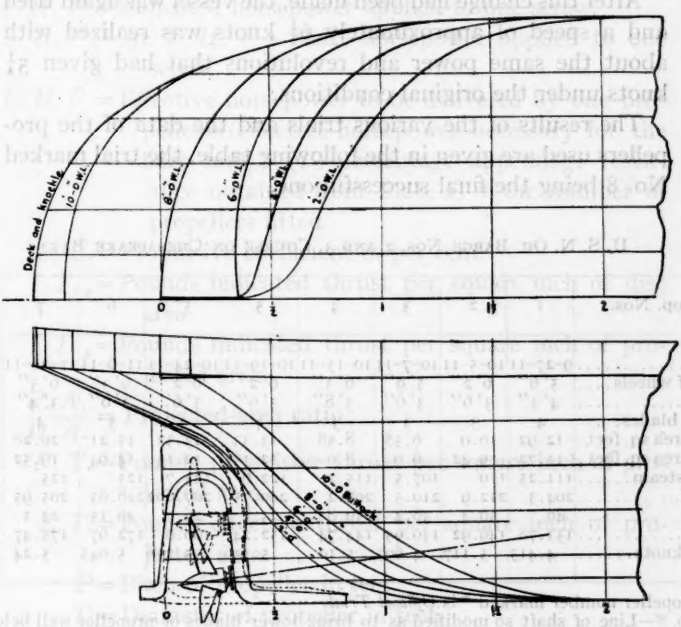


FIG. 11.

Navy Department. The block coefficient of the hull was .9, the after body being very full.

The propeller, as first fitted, was located as shown by the dotted lines in Fig. 11, the hull lines shown being those of the actual vessel.

The contract speed of the barge was six knots, but, although a series of seven propellers was tried, the maximum speed obtained was only approximately $5\frac{1}{4}$ knots.

After thoroughly considering the conditions, it was determined that the best chance of success, at the least expense, was offered by relocating the propeller so that a better flow of water from forward to it would occur. This idea was adopted, and the propeller was located as shown by the full lines of Fig. 11, the shaft being given a very heavy inclination.

After this change had been made, the vessel was again tried and a speed of approximately $6\frac{1}{4}$ knots was realized with about the same power and revolutions that had given $5\frac{1}{4}$ knots under the original conditions.

The results of the various trials and the data of the propellers used are given in the following table, the trial marked No. 8 being the final successful one.

U. S. N. Oil Barge Nos. 2 and 3, Course on Chesapeake Bay.

Prop. Nos.	1	2	3	4	5	5*	6	7	8
Date	9-27-11	10-5-11	10-7-11	10-13-11	10-19-11	10-24-11	11-6-11	12-20-11	5-8-12
Dia. of wheels	5′6″	6′2″	5′9″	6′3″	6′2″	6′2″	6′3″	6′3″	6′9″
Pitch	4′3″	3′6″	3′6″	3′8″	3′6″	3′6″	3′0″	3′4″	3′6″
No. of blades	4	3	3	3	4	4	3	4	3
Dev. area sq. feet.	12.92	10.0	6.58	8.58	13.33	13.33	15.21	20.28	14
Proj. area sq. feet.	11.72	9.42	6.0	8.0	12.13	12.13	14.64	19.52	13.2
Aver. steam	111.25	110	107.5	115.2	122.6	123.7	125	125	129.53
R.P.M.	204.3	212.9	210.5	208.2	200.75	207.602	228.65	205.95	206.27
Slip %	49	30.4	36.4	30.8	23.8	26.7	26.15	22.5	12.56
I.H.P.	133.78	149.92	130.95	144.74	152.24	160.0	172.67	175.47	177.64
Speed knots	4.415	5.117	4.695	5.10	5.136	5.258	5.045	5.24	6.235

Propeller number marked * is *Official Trial*.

No. 8—Line of shaft so modified as to bring lower blades of propeller well below keel of vessel.

Choice of Charts to Use in Design and Method of Using.

Chart 5. To be used when tank curves of *E. H. P.* are available for use in design.

Chart 6. To be used where estimated *I. H. P.* and speed are available, there being no model-tank curves at hand.

Chart 7. To be used when tip speed is 5,000 feet per minute and lower.

Curves of *P. C.* on Plates 3, 5, and 7. These values are for blades of bronze or equivalent metal, as thin as possible, machined to pitch and highly polished. Should the blades be less carefully finished, the *I. H. P.* obtained by the charts should be slightly increased, especially for Chart 5.

NOMENCLATURE USED IN FORMULAS.

I. H. P. = Indicated horsepower applied to one screw.

S. H. P. = .92 *I. H. P.* = Shaft horsepower applied to one screw.

E. H. P. = Effective horsepower to be delivered by one propeller = (effective horsepower necessary for the bare hull) × (1 + percentage appendage resistance obtained from Plate 2) ÷ the number of propellers fitted.

P. C. = Propulsive coefficient in per cent.

$I.T._D$ = Pounds indicated thrust per square inch of disc area.

$I.T._p$ = Pounds indicated thrust per square inch of projected area.

$\frac{P.A.}{D.A.}$ = Projected-area ratio.

P. T. = Pounds propulsive thrust per square inch of projected area.

E. T. = Pounds effective thrust per square inch of projected area.

P = Pitch of propeller in feet.

D = Diameter of propeller in feet.

s = Apparent slip of propeller expressed as a decimal.

S = Speed of ship in knots per hour.
$T.S.$ = Tip speed of propeller in feet per minute.
R = Revolutions of propeller per minute.

$$I.T._D = \frac{I.H.P. \times 33{,}000}{P \times R \times \frac{1}{4}\pi D^2 \times 144} = \frac{S.H.P. \times 33{,}000}{P \times R \times \frac{1}{4}\pi D^2 \times 144 \times .92} \quad (1)$$

$$I.T._p = \frac{I.T._D}{P.A. \div D.A.} \quad (2)$$

$$P.C. = \frac{E.H.P.}{I.H.P.} = \frac{E.H.P. \times .92}{S.H.P.} \quad (3)$$

$$P.T. = I.T._p \times P.C. \quad (4)$$

$$E.T. = \frac{P.T.}{1-s} = \frac{E.H.P. \times 33{,}000}{101.33 \times S \times \pi \frac{D^2}{4} \times 144 \times \frac{P.A.}{D.A.}} \quad (5)$$

$$D = \sqrt{\frac{291.8 \times I.H.P.}{P \times R \times I.T._D}} \quad (6) \qquad 1 - s = \frac{P.T.}{E.T.} \quad (7)$$

$$s = 1 - \frac{P.T.}{E.T.} \quad (8)$$

$$P \times R = \frac{101.33 \times S}{1-s} = \frac{101.33 S \times E.T.}{P.T.} \quad (9)$$

$$I.H.P. = \frac{E.H.P.}{P.C.} \quad (10)$$

$$I.T._D = I.T._p \times \frac{P.A.}{D.A.} = \frac{P.T.}{P.C.} \times \frac{P.A.}{D.A.} \quad (11)$$

Substituting in (6) we have

$$D = \sqrt{\frac{291.8 \times E.H.P. \times P.T. \times P.C.}{P.C. \times 101.33 \times S \times E.T. \times P.T. \times \frac{P.A.}{D.A.}}}$$

$$= \sqrt{\frac{2.88 \times E.H.P.}{\frac{P.A.}{D.A.} \times S \times E.T.}} \quad (12)$$

$$R = \frac{T.S.}{\pi D.} \quad (13)$$

$$P = \frac{P \times R \times \pi D.}{T.\,S.} = \frac{101.33 \times S \times E.\,T. \times \pi D.}{T.\,S. \times P.\,T.} \qquad (14)$$

The work as laid down is for three-bladed propellers. Should a four-bladed screw be desired for special conditions under which the propeller is to operate, multiply the *E. H. P.* of the hull (all appendages) by .865, and use the product as the *E. H. P.* in the calculations. Multiply the projected area of the propeller, thus obtained by 4/3 and the result will be the projected area of the desired four-bladed wheel.

Should a two-bladed wheel be desired divide the *E. H. P.* by .75 and take 2/3 of the projected area of the basic three-bladed propeller obtained by the calculations as the projected area of the new two-bladed propeller.

The efficiency of the four-bladed wheel will probably be slightly less, and that of the two-bladed one slightly greater than that of the three-bladed wheel, but the two-bladed wheel would be unsatisfactory for use in rough water where the vessel would be subjected to heavy rolling and pitching.

Propellers of Reduced Diameter.

In many cases of design, after computing a propeller to absorb a certain amount of power at a given number of revolutions, it will be found that, due to necessary restrictions the deduced diameter cannot be carried, and it becomes necessary to calculate a secondary screw of less diameter, to absorb the designed power without exceeding the designed number of revolutions.

An investigation of the performances of propellers where the diameters after preliminary trials have been slightly reduced, accompanied by a slight reduction in pitch, shows, by comparing the performance before reduction with that after the reduction, that where equal powers are applied to the screws, the

$$\frac{\text{Revolutions after reduction}}{\text{Revolutions before reduction}} = \sqrt[2]{\frac{\text{Diam. before reduction}}{\text{Diam. after reduction}}}$$

or

$$\frac{R_2}{R_1} = \left(\frac{D_1}{D_2}\right)^{\frac{1}{2}}$$

and that

$$\frac{\text{Apparent slip after reduction}}{\text{Apparent slip before reduction}} = \frac{s_2}{s_1} = \left(\frac{D_1}{D_2}\right)^4$$

Now, if the primary screw (before reduction) has a different tip speed than the secondary screw (after reduction), when absorbing equal powers, the tip speeds will have the following ratio:

$$\frac{T.S._2}{T.S._1} = A = \frac{\pi R_2 D_2}{\pi R_1 D_1} = \frac{D_2}{D_1} \times \left(\frac{D_1}{D_2}\right)^{\frac{1}{2}} = \left(\frac{D_2}{D_1}\right)^{\frac{1}{2}}$$
$$\therefore D_2 = D_1 \times A^2$$

and

$$A = A^2 \times \frac{R_2}{R_1} \quad \therefore R_2 = R_1 \div A.$$

Assuming that the total projected areas are the same in the two cases, which can be done safely, as the primary screws under discussion were narrow tipped, so that the reduction of area due to the reduction in diameter was very small; and, as a slight correction has been allowed in the revolution and slip exponents to allow for the slight increase in the area of the secondary screw necessary to bring it equal to that of the primary one, the equation for projected-area ratios takes the following form:

$$\frac{P.A._2}{D.A._2} : \frac{P.A._1}{D.A._1} :: D_1^2 : D_2^2$$

$$\frac{P.A._2}{D.A._2} = \frac{P.A._1}{D.A._1} \times \left(\frac{D_1}{D_2}\right)^2 = \frac{P.A._1}{D.A._1} \times \frac{1}{A^4}$$

The apparent slip and the pitch relations of the two propellers will be expressed by

$$\frac{s_2}{s_1} = \left(\frac{D_1}{D_2}\right)^4 = \frac{1}{A^8} \therefore s_2 = \frac{s_1}{A^8} \qquad \text{and} \qquad P_2 = P_1 \times A.$$

Rules for Entering the Plates of Design.

1. Should the diameter of the propeller be limited, not to exceed a certain value D, and the designed revolutions of the propeller to have a certain value R, the allowable maximum tip speed with this limiting diameter and revolutions will be $T.S. = \pi D \times R$. Enter Plate 5, Plate 6, or for low tip speeds, Plate 7, with this tip speed, and use the values of $E.T.$, $P.T.$, and $P.C.$ on Plates 5 and 7, or $I.T.$ per square inch disc area and per cent. apparent slip, Plate 6, on the same ordinate as the limiting tip speed, in obtaining the trial propeller. Should this trial propeller be found to have fewer revolutions and greater diameter than the limiting diameter and revolutions, solve for a series of propellers, very gradually increasing the tip-speed values, until the desired propeller is obtained. Should the diameter be less and the revolutions greater than desired, solve in a similar manner but with the tip speeds gradually reducing. Should the diameter be much greater and the revolutions much less than the limit for the limiting tip speed, the problem becomes one of "Reduced diameter," similar to problems 1 and 2, given later.

Steam launches, colliers and other vessels whose speed does not exceed 14 to 15 knots, and where the limit of diameter is small or the revolutions of the propellers low, not exceeding 100, usually produce problems of "Reduced diameters."

Rule 2. For vessel of moderate speeds, from 15 to 21 knots approximately, and with revolutions and diameter limits such that the tip speeds are not less than approximately 5,000 feet, enter Plate 5 or 6, with a value of $\frac{P.A.}{D.A.} = .3$ to obtain the initial screw, increasing $\frac{P.A.}{D.A.}$, or decreasing it in the series of propellers according to whether the revolutions of the initial screw are less or greater than those desired.

Rule 3. For destroyers, scouts and torpedo boats where the tip speed varies from about 7,400 to 8,500 feet, enter the plates of design with $\frac{P.A.}{D.A.} = .35$ for the initial propeller.

Rule 4. For turbine-propelled battleships and other large, heavy vessels of approximately 21 to 22 knots speed, enter the plates with $\frac{P.A.}{D.A.} = .50$ to .55 for the initial propeller, usually following with increased values of $\frac{P.A.}{D.A.}$, or solving for reduced diameters.

Rule 5. For scouts and destroyers of high speed, enter the plates of design with $\frac{P.A.}{D.A.} = .55$ to .60 for the initial screw. In the case of four-shaft vessels, the wing screws may be designed separately from the inner ones on account of the difference in conditions under which they operate. Usually, however, for initial economy and reduction in the number of spares, they are built alike as the differences in the deduced screws will be found very small.

Rule 6. Use Plate 5 for design when the *E. H. P.* curve of the vessel is available; when it is not, and the estimated *I. H. P.* and speed are given, use Plate 6. For tip speeds of 5,000 feet and lower, use Plate 7.

PROBLEMS FOR THREE-BLADED PROPELLERS.

Problem 1. Limited diameter, low tip speed.—Required a propeller for a launch whose nominal block coefficient is .3638 and whose ratio of immersed beam to length on load-water line, $\frac{B}{L.W.L.}$ is .2625. The estimated speed is $6\frac{1}{2}$ knots, revolutions 500 per minute, and *I. H. P.* 21.74. There is one propeller, working directly in wake of, and close to, the stern post. There are no *E. H. P.* curves available and the *I. H. P.* is an estimate. Maximum diameter allowable = 24 inches.

In this problem, the *I. H. P.* is only an estimate of the power required, and, as no *E. H. P.* curve is furnished, the question of propulsive coefficient is entirely eliminated from the problem. If the estimate of power for speed is too low, the propeller obtained will be too small, the slip will be higher than estimated and the boat may fall below speed. Should

the estimate be too large, the propeller will be larger and have more surface than is required, the revolutions will decrease, the slip will decrease, and the power required per revolution will increase. In cases of launches and, in fact, in all cases in general, it is preferable to make the screw slightly larger than is absolutely necessary for trial conditions, as it will be all the better for heavier displacements than that of the trials.

The tip speed of this problem being very low, and the diameter very small, it is probable that the propeller obtained directly from the calculations, as necessary to absorb the designed power at the required revolutions, will be of considerably larger diameter than can be carried, and therefore the tip speed for this basic propeller will be higher than that of the actual propeller.

To ascertain this fact we solve the problem first for a propeller having the same tip speed as the final screw, namely, 3,142 feet per minute.

Having corrected the block coefficient by Plate 1, for variation of $\frac{B}{L.W.L.}$ from the standard value, and for the position of the screw, using the intersection of line Z, Plate 1, with the line passing through the abscissa value unity and the plotted position of the nominal block coefficient on Plate 1, we find that standard block coefficient curve of apparent slip to use in the computation is .715.

Turning to Plate 7, and passing an ordinate through the 3,142 point on the tip-speed curve, it will be found to intersect the base at an abscissa value of $I.\ T._D = .56$, the $P.\ T.$ curve at 2.08 pounds, and the $E.\ T.$ curve for $B.\ C. = .715$ at 2.225.

The value 1—slip $= 1 - s_1 = \frac{P.\ T.}{E.\ T.} = .9348.$

The speed being $6\frac{1}{2}$ knots, $101.33 \times S = 101.33 \times 6\frac{1}{2} = 658.65.$

Pitch $\times$ Revolutions $= P \times R = \frac{101.33\ S}{1 - s_1} = \frac{658.65}{.9348} = 704.6$

The $I.\ H.\ P. = 21.74$ by the estimate.

$$\text{Diam.}_1 = \sqrt{\frac{291.8 I.H.P.}{P \times R \times I.T._D}} = \sqrt{\frac{291.8 \times 21.74}{704.6}} = 4.01 \text{ feet.}$$

$$\text{Revolutions}_1 = \frac{\text{Tip Speed}}{\pi D} = \frac{3{,}142}{\pi \times 4.01} = 249.4$$

$$P_1 = \frac{P_1 \times R_1 \times \pi D_1}{R_1 \times \pi \times D_1} = \frac{P_1 \times R_1 \times \pi D_1}{T.S.} =$$

$$\frac{658.65 \times \pi \times 4.01}{3{,}142} = 2.825 \text{ feet.}$$

As is seen, the diameter of the propeller above is much larger than can be carried by the boat and the revolutions are far below those desired. In order to obtain the proper propeller we proceed as shown in columns 2, 3, 4 and 5, column 2 being taken as the example to explain in the following.

The tip speed of the theoretically necessary blade is assumed as 3,300 feet.

The ratio $A = \frac{3{,}142}{3{,}300}$, is the ratio existing between the theoretical full-length blade-tip speed and the tip speed of the reduced propeller, and equals .9521.

The reduced propeller and the theoretical propeller, in order to hold to the conditions from which the reduction factors are obtained, must have the same total projected area, so that the projected-area ratios will vary inversely as the squares of the diameters or inversely as A^4, so that $\frac{P.A.}{D.A.}$ of final propeller will equal $\frac{P.A.}{D.A.}$ of theoretical propeller $\div A^4$.

As pointed out previously, experiment has shown that

$$D = D_1 \times A^2, \quad R = R_1 \div A, \quad s = s_1 \div A^8, \quad P = P_1 \times A$$

and

$$\frac{P.A.}{D.A.} = \frac{P.A._1}{D.A._1} \div A^4.$$

Passing the ordinate on Plate 7, through the tip-speed curve at a value of 3,300 feet, the $I.T._D$ and the $P.T.$ for

.715 will be found to be .63 pounds and 2.35 pounds while the value $E.\ T. = 2.51$.

The slip for the theoretical screw being $1 - \frac{P.\ T.}{E.\ T.} = .0638$, s, that for the final screw will be $s = \frac{s_1}{A^3} = \frac{.0638}{.6755} = .09445$, and from this value of s, we obtain the value of $P \times R = P_1 \times R_1$.

As before, $S = 6\frac{1}{2}$, $101.33S = 101.33 \times 6\frac{1}{2} = 658.65$.

$$P \times R = \frac{658.65}{1 - .09445} = P_1 \times R_1 = 727.3.$$

$I.\ H.\ P. = 21.74$.

$$D_1 = \sqrt{\frac{291.8\ I.H.P.}{P \times R \times I.T._D}} = \sqrt{\frac{291.8 \times 21.74}{727.3 \times .63}} = 3.721 \text{ ft.}$$

$$\text{Revolutions} = \frac{\text{Tip Speed}}{\pi D_1} = \frac{3{,}300}{\pi \times 3.721} = 282.3$$

$$\text{Pitch} = P_1 = \frac{P_1 \times R_1 \times \pi D_1}{R_1 \times \pi D_1} = \frac{P_1 \times R_1 \times \pi D_1}{T.S._1} = \frac{727.3 \times \pi \times 3.721}{3{,}300} = 2.576 \text{ ft.}$$

The diameter obtained is too large. The resultant diameter for the final propeller of 3,142 feet tip speed will be $3.721 \times A^2 = 3.373$ feet.

The pitch of the final screw will be that of the theoretical one, $2.576 \times A = 2.453$ feet.

The projected-area ratio will be—

$$.2 \div A^4 = .2433.$$

The revolutions of the reduced propeller will be $282.3 \div A = 282.3 \div .9521 = 296.5$.

This propeller of reduced diameter has therefore the following characteristics:

Diameter = 3.373 feet.
Pitch = 2.453 feet.
$\frac{P.\ A.}{D.\ A.} = .2433$.
Revolutions = 296.5.

This propeller is too low in revolutions, so it is necessary to obtain several others of different assumed theoretical tip speeds as indicated by columns 3, 4 and 5, and lay down curves of Pitch-Diameter, Revolutions-Diameter, $\frac{P.A.}{D.A.}$-Diameter, using the reduced diameter as abscissa, from which curves the propeller of the desired revolutions can be selected.

PROBLEM 1.

Column	(1)	(2)	(3)	(4)	(5)
$T.S.$	3,142	3,142	3,142	3,142	3,142
$T.S._1$	3,142	3,300	3,500	3,700	4,000
$P.A._1 \div D.A._1$	.2	.2	.2	.2	.2
$I.T._D$	.56	.63	.74	.86	1.07
$P.T.$	2.08	2.35	2.74	3.17	3.92
$E.T.$	2.225	2.51	2.92	3.37	4.16
$\frac{P.T.}{E.T.} = 1 - s_1$	.9348	.9362	.9383	.9406	.9423
s_1	.0652	.0638	.0617	.0594	.0577
A	1	.9521	.8977	.8492	.7855
A^3	1	.9065	.8059	.7211	.6170
A^4	1	.8219	.6495	.5200	.3807
A^8	1	.6755	.4218	.2704	.1449
$s = \frac{s_1}{A^8}$	.0652	.09445	.1463	.2196	.3981
S	6.5	6.5	6.5	6.5	6.5
$101.33\ S$	658.65	658.65	658.65	658.65	658.65
$P \times R = \frac{101.33S}{1-s}$	704.6	727.3	771.5	844.0	1638.8
$I.H.P.$	21.74	21.74	21.74	21.74	21.74
$D_1 = \sqrt{\frac{291.8\ I.H.P.}{P \times R \times I.T._D}}$	4.01	3.721	3.333	2.956	1.902 feet
$D = D_1 \times A^3$	4.01	3.373	2.686	2.132	1.174 feet
$P_1 = \frac{P \times R \times \pi D_1}{T.S._1}$	2.825	2.576	2.308	2.119	2.448 feet
$P = P_1 \times A$	2.825	2.453	2.072	1.799	1.923 feet
$R_1 = \frac{T.S._1}{\pi D_1}$	249.4	282.3	334.4	398.4	669.4
$R = R_1 \div A$	249.4	296.5	372.5	469.1	852.2
$\frac{P.A.}{D.A.} = \frac{P.A._1}{D.A._1} \div A^4$	.2	.2433	.308	.3846	.5253

Problem 2.—Cargo steamer to steam constantly at 14 knots; twin screws.—Revolutions for that speed to be about 90 per minute. *E. H. P.* of bare hull for 14 knots = 3,500. Nominal block coefficient = .78. $\frac{B}{L.W.L.} = .127$. Coeffi-

cient of immersed midship section = .98. Total power of engine 4,050 per engine with revolutions not exceeding 98. Required propeller and *I. H. P.* for the 14-knot condition and for maximum power.

The vessel plots in Plate 1, at point marked *A*, and the corrective line through *A* cuts *X* at a block coefficient value of .665, which, as the standard coefficient of immersed midship section at *C* agrees with that of the vessel under consideration, is the block coefficient line of *E. T.* to use. The plates of design to use are Plates 7 and 5, as the model-tank curves of *E. H. P.* are available.

The maximum diameter allowable is 16 feet 6 inches, imposed on the propeller by the necessity of keeping a certain depth of immersion of blade tips and a sufficient clearance between the blade tips and the hull. At 90 revolutions the tip speed for this diameter will be 4,665.24 feet per minute. This tip speed is low, and it may be found that the problem is one of reduced diameters, as in Problem 1.

As the *E. H. P.* given is that for the bare hull, it must be corrected for the increased resistance due to the appendages, which, by referring to Plate 2, for a $\frac{B}{L.W.L.}$ of .127, we see to be approximately 13 per cent. Therefore the total $E.\ H.\ P. = 3{,}500 \times 1.13 = 3{,}955$.

The vessel is to have twin screws, hence the *E. H. P.* to be delivered by each propeller at 14 knots $= \frac{3{,}955}{2} = 1{,}978$.

As the tip speed is low, it may be found that the problem becomes similar to Problem 1, therefore the form of computation is prepared as in Problem 1.

Should it be estimated that the designed power of the engines for 14 knots is very close to the maximum possible power, the propeller may be calculated for the 14-knot condition, as in the first column of the tabulated form of computation for Problem 2. Should the estimated overload be large, as it is in this problem, the overload condition should be used in the design, as in the second and following columns of the tabulation.

Entering Plates 7 and 5, with tip speeds of 4,665 and 5,080 feet per minute, which are the tip speeds corresponding to the allowed maximum diameter and the desired revolutions, it is seen from Plate 7 that for 4,665 feet tip speed the *P. T.*, *E. T.*, and *P. C.* are 5.9 pounds, 6.4 pounds, and 72.7 per cent. For a tip speed of 5,080 feet, the corresponding values from Plate 5 are 7.14 pounds, 7.85 pounds, 71.2 per cent., and .2075. In both cases, the values of *E. T.* are taken for the standard block .665.

From the *E. H. P.* curves, corrected for appendage resistance by Plate 2, the speed corresponding to 8,100 *H. P.* = 4,050×2 is 15.55 knots, the *E.H.P.* for one propeller being $\frac{8,100}{2} \times .712 = 2,884$.

Solving the first two columns by using this primary data, the P_1, D_1 and R_1 are found to be

	For 4,665 feet *T. S.*	For 5,080 feet *T. S.*
D_1	17.63 feet	18.11 feet
P_1	18.28 feet	19.40 feet
R_1	84.21	89.3

It is evident from the above results that the propeller obtained in the first column is of too small diameter and pitch to efficiently absorb the total estimated power of the engines, and to hold the revolutions at that power down to the desired revolutions. Column 2 is therefore taken as the basic screw and columns 3, 4 and 5 are solved, according to the method of solving for "reduced diameters."

The method to be followed in this solution has been fully explained in Problem 1, and will not be described here.

Having obtained the secondary reduced propellers in columns 3, 4 and 5, the results may be shown graphically as curves, using reduced diameters as abscissæ, and for ordinates using the corresponding pitch, revolution and projected-area ratio values, and the resultant propeller for 16.5 diameter be taken directly from these curves.

The characteristics will be found to be approximately

$D = 16$ feet 6 inches — Revolutions $= 98$

$P = 17$ feet 9.6 inches — $H.P. = 8{,}100$

$\frac{P.A.}{D.A.} = .25$ — Slip $= 12$ per cent

Speed $= 15.15$

PROBLEM 2—*A-D*.

	For continuous cruising at 14 knots.	For overload—8,100 H. P., 98 revolutions. Design on Plate 5—line *A-D*.			
$T.S.$	4,665.24	5,080	5,080	5,080	5,080
$T.S.$	4,665.24	5,080	5,200	5,400	5,600
$A = \frac{T.S.}{T.S._1} = \frac{R\pi D}{R_1\pi D_1}$	1	1	.9769	.9407	.9071
A^2	1	1	.9544	.885	.82291
A^4	1	1	.9108	.7832	.6787
A^6	1	1	.8296	.6134	.4606
$\frac{P.A._1}{D.A._1}$	.2	.2075	.215	.225	.2375
$P.T._1$	5.9	7.14	7.22	7.4	7.55
$E.T._1$	6.4	7.85	7.98	8.26	8.39
$1-s_1 = \frac{P.T._1}{E.T._1}$	.9218	.9096	.9047	.8959	.9000
$s_1 = 1 - \frac{P.T._1}{E.T._1}$	.0782	.0904	.0953	.1041	.1000
$s = \frac{s_1}{A^6}$	.0782	.0904	.1149	.1698	.2181
S_1	14	15.55	15.54	15.52	15.5
$101.33 \times S_1$	1,418.62	1,575.6	1,574.7	1,572.6	1,570.6
$P \times R = \frac{101.33 \times S_1}{1-s}$	1,539	1,732.2	1,740.4	1,755.3	1,745.1
$S_1 = \frac{(1-s)\,P \times R}{101.33}$	14	15.55	15.25	14.38	13.48
$E.H.P.$	1,978	2,884	2,880	2,872	2,855
$D_1 = \sqrt{\frac{2.88 \times E.H.P.}{\frac{P.A._1}{D.A.} \times S_1 \times E.T._1}}$	17.63	18.11	17.64	16.93	16.32
$D = D_1 \times A^2$	17.63	18.11	16.83	14.98	13.43
$P_1 = P \times R_1 \times \pi D_1 \div T.S.$	18.28	19.40	18.55	17.29	15.98
$P = P_1 \times A$	18.28	19.40	18.12	16.26	14.5
R_1	84.21	89.3	93.83	101.5	109.3
$R = R_1 \div A$	84.21	89.3	96.95	107.9	120.5
$\frac{P.A.}{D.A.} = \frac{P.A._1}{D.A.} \div A^4$	.2	.2075	.2361	.2873	.3508
$I.H.P. = E.H.P. \div P$ Coef.	2,721	4,050	4,050	4,050	4,050

A successful propeller for a vessel very similar in form to the vessel under consideration and for very similar conditions had the following characteristics and results.

$D = 16$ feet 6 inches — $H.P. = 8{,}400$

$P = 18$ feet 0 inches — $S = 15.5$

$\frac{P.A.}{D.A.} = .278$ — Revolutions $= 98.2$

Slip $= 11.14$

In order to obtain the approximate revolutions and power of this propeller at 14 knots, assume that the speed will vary as the revolutions and that the power will vary as the cube of the speeds:

$$I.H.P. : I.H.P._2 :: S_1^3 : S_2^3$$

$$R_1 : R_2 :: S_1 : S_2$$

$$I.H.P. : 8{,}100 :: 14^3 : 15.15^3$$

$$R : 98 :: 14 : 15.15$$

$$I.H.P. = \frac{8{,}100 \times 14^3}{15.15} = 6{,}392$$

$$R = \frac{98 \times 14}{15.15} = 90.56$$

Problem 3.—Vessel of nominal block coefficient of .65, $\frac{B}{L.W.L.} = .196$. Coefficient of immersed midship section $= .94$. Estimated $I.H.P. = 12{,}000$ on two propellers. Revolutions of main engines at that power $= 116$. Estimated speed 17 knots. Required propeller for the estimated conditions. Diameter of propeller not to exceed 17 feet.

Tip speed at this diameter and 116 revolutions $= 6{,}195$. Use Plate 1 for block correction, and Plate 6 for propeller design, entering Plate 6 with tip speed allowable, namely, 6,195 feet per minute. Corrected block $= .67$.

$T.S.$	$\frac{P.A.}{D.A.}$	$I.T.$	$H.P.$	S	s	$101.33\,S$	$\frac{P \times R = 101.33S}{1-s}$	D	P	R
6,195	.273	3.22	6,000	17	.110	1,722.6	1,935.2	16.76	16.45	117.6
5,830	.25	2.82	6,000	17	.102	1,722.6	1,918.3	17.99	18.6	103.1
6,080	.265	3.10	6,000	17	.107	1,722.6	1,929.0	17.11	17.05	113.1
6,160	.270	3.20	6,000	17	.109	1,722.6	1,933.3	16.82	116.59	116.6

By plotting—

$$\text{Diameter} = 16 \text{ feet } 10\tfrac{1}{2} \text{ inches}$$
$$\text{Pitch} = 16 \text{ feet } 7\tfrac{3}{4} \text{ inches}$$
$$\frac{P.A.}{D.A.} = .267$$
$$\text{Revolutions} = 116$$

Suppose the limit of diameter to be 16 feet 9 inches, and it is desired to have a propeller of higher $P \div D$ value than is obtained directly as above. In such case, the problem can be extended as one of "reduced diameter."

Problem 4.—Battleship to make 21 knots. Total *E. H. P.* (all appendages) 15,200; twin screw; revolutions 125 per minute. Block coefficient .6. $\frac{B}{L.W.L.}$ and coefficient of immersed midship section are standard, as per Plate 1. Required propeller and *I. H. P.*

Use $\frac{P.A.}{D.A.} = .3$ for initial propeller. Design with Plate 5.

As, in such vessels as the above, a diameter of nearly 19 feet can usually be carried, the propeller is obtained at once without having recourse to the "reduced diameter" condition. The propeller of $.32 \frac{P.A.}{D.A.}$ may be taken, or the results may be plotted on diameter as abscissa, and the final propeller be taken from the curves so obtained. (See page 622.)

Problem 5.—(High power, 2, 3 and 4 propellers.) Battleship, 2, 3 or 4 propellers, power divided equally between the shafts. Hull of standard block ratios, block coefficient = .64. Designed speed 22.5 knots. *E. H. P.* (all appendages) for this speed 25,000. *E. H. P.* (all appendages) for 15 knots = 5,250. Required *I. H. P.*, *S. H. P.*, propellers and revolutions for full power with equal division of power on shafts, and determine power and revolutions for 15 knots speed.

Block coefficient for center screw, from Plate 1, the dead wood being cut away = .668. (Use Line *Y*, Plate 1.)

$\frac{P.\ A.}{D.\ A.}$	T. S. Plate 5	P. T. Plate 5	E. T. Plate 5	$1-s=\frac{P.\ T.}{E.\ T.}$	S	$101.33S$	$P\times R=\frac{101.33\ S}{1-s}$	E. H. P. one screw	$D=\sqrt{\frac{2.88\ E.\ H.\ P.}{\frac{P.\ A.}{D.\ A.}\times S\times E.\ T.}}$	$P=\frac{P\times R\times\pi D}{T.\ S.}$	$R=\frac{T.\ S.}{\pi D}$	P. C. Plate 5	$\frac{I.\ H.\ P.=\frac{E.\ H.\ P.}{P.\ C.}\times 2}{\text{for two screws}}$
.3	6,710	8.42	9.7	.868	21	2,127.93	2,451.5	7,600	18.45	21.75	112.8	.67	22,687.
.31	6,900	8.55	9.86	.8671	21	2,127.93	2,454.1	7,600	18.47	20.63	118.9	.662	22,961
.32	7,080	8.65	10.00	.865	21	2,127.93	2,460.0	7,600	18.05	19.70	124.9	.655	23,206
.38	7,260	8.74	10.14	.8619	21	2,127.93	2,468.9	7,600	17.65	18.85	131.0	.649	23,421
.34	7,420	8.82	10.27	.8588	21	2,127.93	2,477.8	7,600	17.28	18.13	136.7	.642	23,676

PROBLEM 4.

P. C. Plate 5	$\frac{P.\ A.}{D.\ A.}$	T. S. Plate 5	P. T. Plate 5	E. T. Plate 5	$\frac{P.\ T.}{E.\ T.}=1-s$	S	$101.33S$	$P\times R=\frac{101.33S\times E.\ T.}{P.\ T.}$	$D_1=\sqrt{\frac{2.88\ E.\ H.\ P.}{\frac{P.\ A.}{D.\ A.}\times S\times E.\ T.}}$	$P_1=\frac{P\times R\times\pi D}{T.\ S.}$	$R_1=\frac{T.\ S.}{\pi D}$
					Two Shafts				*Feet.*	*Feet.*	
.532	.50	10,520	9.32	11.15	.8359	22.5	2,279.93	2,727.5	17.66	14.38	189.6
					Three Shafts—Wing Propellers						
.532	.50	10,520	9.32	11.15	.8359	22.5	2,279.93	2,727.5	14.42	11.74	232.2
					Center Propeller						
.532	.50	10,520	9.32	11.08	.8412	22.5	2,279.93	2,710.3	14.46	11.71	231.5
					Four Shafts—Wing Propellers						
.532	.50	10,520	9.32	11.15	.8359	22.5	2,279.93	2,727.5	12.49	10.17	268.2
					Inboard Propellers						
.532	.50	10,520	9.32	11.08	.8412	22.5	2,279.93	2,710.3	12.52	10.14	267.3

PROBLEM 5.

$E.\ H.\ P.$ on one propeller $= \frac{\text{2-shaft}}{12{,}500}, \frac{\text{3-shaft}}{8{,}334}, \frac{\text{4-shaft}}{6{,}250}$.

$I.H.P. = \frac{25{,}000}{.532} = 47{,}000.$ $S.H.P. = I.H.P. \times .92 = 43{,}240.$

Should we desire revolutions higher than those obtained, solve for higher values of $\frac{P.A.}{D.A.}$ or the ratio $\frac{P}{D}$ can be increased by solving for "Reduced Diameter." The method to be followed in this latter case is as described below, there being so little difference between the wing and center or inner screws, that they may be taken as similar. See page 622.

Suppose it is required that the value $\frac{P}{D} = .9$. Taking .50 $\frac{P.A.}{D.A.}$ on Plate 5, with its corresponding values of $E.\ T.$, $P.\ T.$, $P.\ C.$ and $T.\ S._1$ as the primary data, assume varying values of $T.\ S.$ for the reduced propellers. From the ratio A between the tip speeds the values of the apparent slips for the reduced screws are obtained as $s = \frac{s_1}{A^3}$.

Solving the problem for three shafts, with the reduced propellers, a speed of vessel of 22.5 knots must be realized, therefore $P \times R = P_1 \times R_1 = \frac{101.33 \times S}{1-s} = \frac{101.33 \times 22.5}{1-s}$

and this must equal $\frac{101.33 \times S_1}{1-s_1}$.

Therefore, as $1-s_1 = \frac{P.\ T.}{E.\ T.}$ (values from Plate 5),

$$S_1 = \frac{P \times R \times (1-s_1)}{101.33} = \frac{P_1 \times R_1\ (1-s_1)}{101.33}.$$

The values of S_1 so obtained are used in the equation for the primary diameter,

$$D_1 = \sqrt{\frac{2.88\ E.\ H.\ P.}{\frac{P.A._1}{D.A._1} \times E.\ T. \times S_1}},$$ where $E.H.P. = 8{,}334$ in this case.

$\frac{P.A._1}{D.A.}$	.50	.50	.50	.50	.50
$T.S._1$	10,520	10,520	10,520	10,520	10,520
$T.S.$	9,600	9,800	10,000	10,200	10,400
$A=\frac{T.S.}{T.S._1}$	.9127	.9316	.9505	.9696	.9886
A^2	.8331	.8678	.9036	.9401	.9773
A^4	.6941	.7531	.8164	.8837	.9551
A^8	.4817	.5671	.6666	.7810	.9122
$P.T.$	9.32	9.32	9.32	9.32	9.32
$E.T.$	11.15	11.15	11.15	11.15	11.15
$1-s_1=\frac{P.T.}{E.T.}$	.8359	.8359	.8359	.8359	.8359
$s_1=1-\frac{P.T.}{E.T.}$	.1641	.1641	.1641	.1641	.1641
$s=\frac{s_1}{A^8}$	.3359	.2848	.2417	.2068	.1771
S	22.5	22.5	22.5	22.5	22.5
$101.33\ S$	2,279.93	2,279.93	2,279.93	2,279.93	2,279.93
$\frac{101.33\ S}{1-s}=P\times R$	3,433.1	3,186.4	3,006.7	2,874.4	2,770.6
$S_1=\frac{P\times R\times(1-s_1)}{101.33}$	28.32	26.28	24.8	23.71	22.85
$D_1=\sqrt{\frac{2.88\ E.H.P.}{\frac{P.A.}{D.A._1}\times E.T.\times S_1}}$	12.33	12.8	13.17	13.47	13.72
$D=D_1\times A^2$	10.31	11.15	11.95	12.72	13.43
$P_1=\frac{P\times R\times\pi D_1}{T.S._1}$	12.66	12.2	11.85	11.59	11.38
$P=A\times P_1$	11.55	11.36	11.26	11.24	11.25
$R_1=\frac{T.S._1}{\pi D_1}$	271.1	261.1	253.7	248.	243.5
$R=R_1\div A$	296.5	279.8	266.4	255.3	245.8
$\frac{P.A.}{D.A_1}=\frac{P.A._1}{D.A_1}\div A^4$	.7156	.6589	.6077	.5615	.5195
$P.C.$	.532	.532	.532	.532	.532
$I.H.P.=\frac{E.H.P.}{P.C._1}$	47,000	47,000	47,000	47,000	47,000 on 3 screws
$S.H.P.=H.P.\times.92$	43,240	43,240	43,240	43,240	43,240

To extend this to four shafts, as the tip speeds, $P.T.$, $E.T.$, and $\frac{P.A.}{D.A.}$ are constant, the variables D_1, P, will vary directly as the $\sqrt{E.H.P.}$ absorbed by the propellers under the two conditions, that is of three and four screws, while the revolutions will vary inversely as the diameters:

Thus $D_{1-4}=D_{1-3}\sqrt{\frac{6,250}{8,334}}$, and

$$D_4=D_3\times\sqrt{\frac{6,250}{8,334}}=\begin{Bmatrix}10.31\\11.15\\11.95\\12.72\\13.43\end{Bmatrix}\times\sqrt{\frac{6,250}{8,334}}=\begin{Bmatrix}8.929\\9.656\\10.35\\11.02\\11.63\end{Bmatrix}\text{ feet.}$$

P will vary in the same manner, so that

$$P_4=P_3\times\sqrt{\frac{6,250}{8,334}}=\begin{Bmatrix}11.55\\11.36\\11.26\\11.24\\11.25\end{Bmatrix}\times\sqrt{\frac{6,250}{8,334}}=\begin{Bmatrix}10.01\\9.842\\9.755\\9.731\\9.742\end{Bmatrix}\text{ feet.}$$

The revolutions will vary inversely as the diameters, hence

$$R_4=R_3\times\sqrt{\frac{8,334}{6,250}}=\begin{Bmatrix}296.5\\279.8\\266.4\\255.3\\245.8\end{Bmatrix}\times\sqrt{\frac{8,334}{6,250}}=\begin{Bmatrix}342.5\\323.1\\307.6\\294.8\\283.8\end{Bmatrix}$$

and $\left(\frac{P.A.}{D.A.}\right)_4$ will equal $\left(\frac{P.A.}{D.A.}\right)_3$

To obtain the revolutions and power for 15 knots, as called for by the problem, let two conditions with the three-shaft arrangement be assumed.

1. Full Diameter—Propeller No. 1.

Diameter, = 14.44 feet
Pitch = 11,725 feet
Revolutions = 232.
$\frac{P.A.}{D.A.}$ = .50

2. Reduced diameter—

	Propeller No. 2.	Primary Screw.
Diameter . . .	13.43 feet.	13.72 feet.
Pitch	11.25 feet.	11.38 feet.
Revolutions.	245.8	243.5
$\frac{P.A.}{D.A.}$	.5195	.50

Required revolutions and power for 15 knots with each propeller, Nos. 1 and 2. Use Plates 3 and 4.

Assume that speed will vary approximately as the revolutions, and using the primary propeller of the second case for the preliminary calculations for the second propeller:

$$R_1 = \frac{232 \times 15}{22.5} = 154.67, \text{ and } T.S. = 7{,}017 \text{ feet.}$$

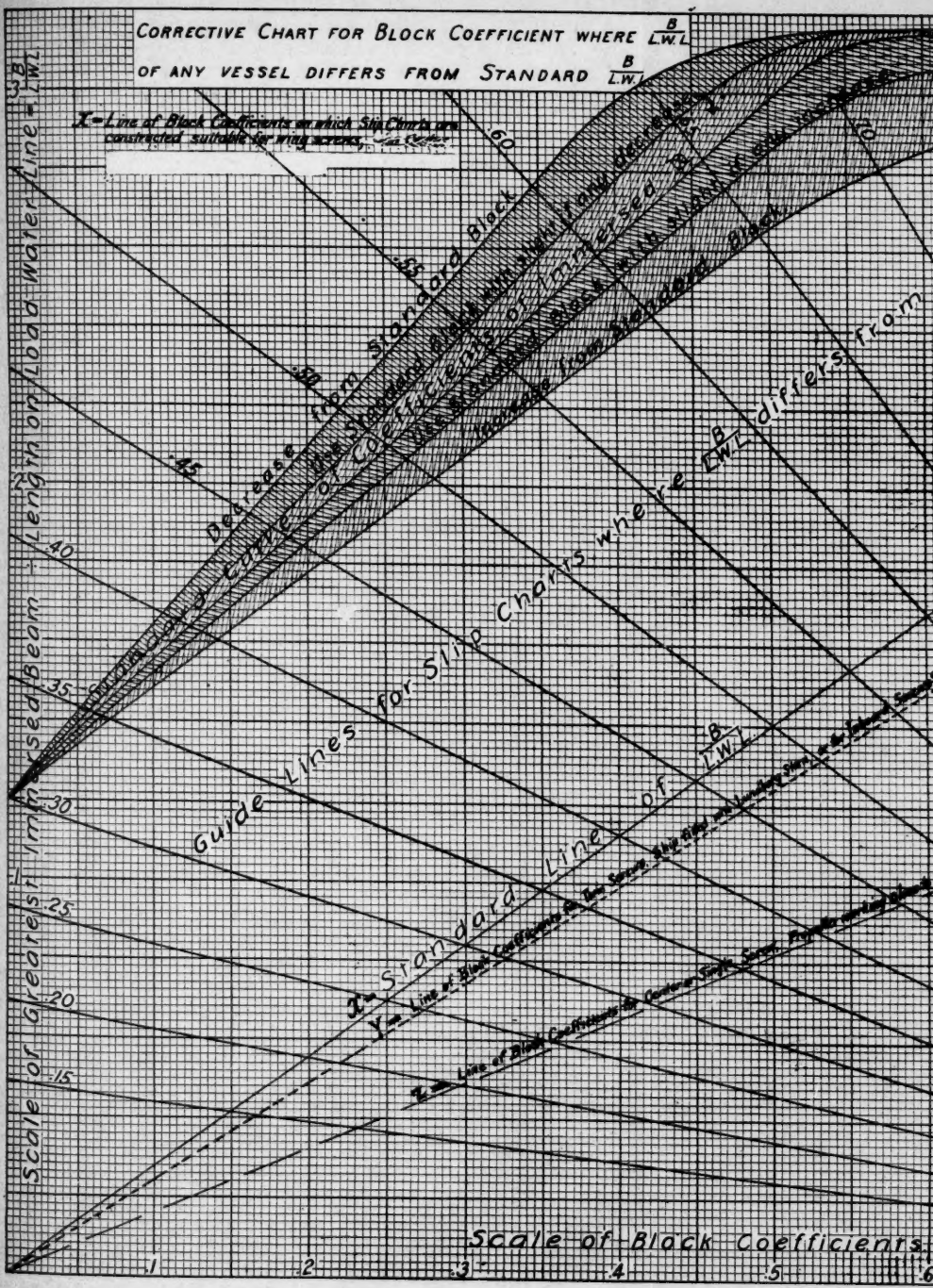
$$R_1 = \frac{243.5 \times 15}{22.5} = 162.33, \text{ and } T.S. = 7{,}118 \text{ feet.}$$

Entering Plate 4, with these tip speeds for *I. T.* per square inch of disc area, and with these *I. T.*'s entering Plate 3 for *P. C.*s, both *T. S.* and *P. C.* being those for the projected-area ratios of the two propellers, using the primary propeller in the second case, the following primary data is tabulated:

$\frac{P.A.}{D.A.}$	*T. S.* Plate 4	*I. T.* Plate 4	*P. C.* Plate 5	$R = \frac{T.S.}{\pi D}$	$I.T. \times \frac{\pi D^2}{4} \times 144 \times P. \times R. \div 33000 = I.H.P.$	$E.H.P. = I.H.P. \times P.C.$	Total $E.H.P. = 3 \times E.H.P.$	Total $S.H.P. = 3 \times I.H.P. \times .92$
.50	7,017	2.98	.609	154.7	3,862	2,352	7,056	10,659
.50	7,118	3.08	.610	162.33	3,661	2,233	6,699	9,924

As the effective horsepower for 15 knots is only 5,250, the above results are high and the calculations must be carried down through a lower range of tip speeds and *I. T.*s until the 15-knot condition is included in the range. The results may then be plotted as curves of *E. H. P.* and *S. H. P.* on revolutions as abscissæ, correcting the revolutions for the reduced screw by dividing those of the primary screw by *A* (page 624), and the 15-knot condition be taken directly from the curves.

The same method is used when the model tank *E. H. P.* curves are available, for laying down curves of estimated *H. P.* revolutions, *S. H. P.* revolutions, and speed revolutions by obtaining by means of Plates 3 and 4 for different tip speeds, the corresponding *I. H. P.*s, *S. H. P.*s and revolutions and *P. C.*s, and with the resultant *E. H. P.*s, taking the speeds corresponding from the *E. H. P.* speed curve.


CORRECTIVE CHART FOR BLOCK COEFFICIENT WHERE B/L.W.L OF ANY VESSEL DIFFERS FROM STANDARD B/L.W.L
X = Line of Block Coefficients on which Slip Charts are constructed suitable for ship screws.
Length of load water line = L.W.L
Scale for Greatest Immersed Beam = B
Guide Lines for Slip Charts where B/L.W.L differs from Standard Line of B/L.W.L
Scale of Block Coefficients
.1
.2
.3
.4
.5
.6
.15
.20
.25
.30
.35
.40
.45
.50
.55
.60
.70

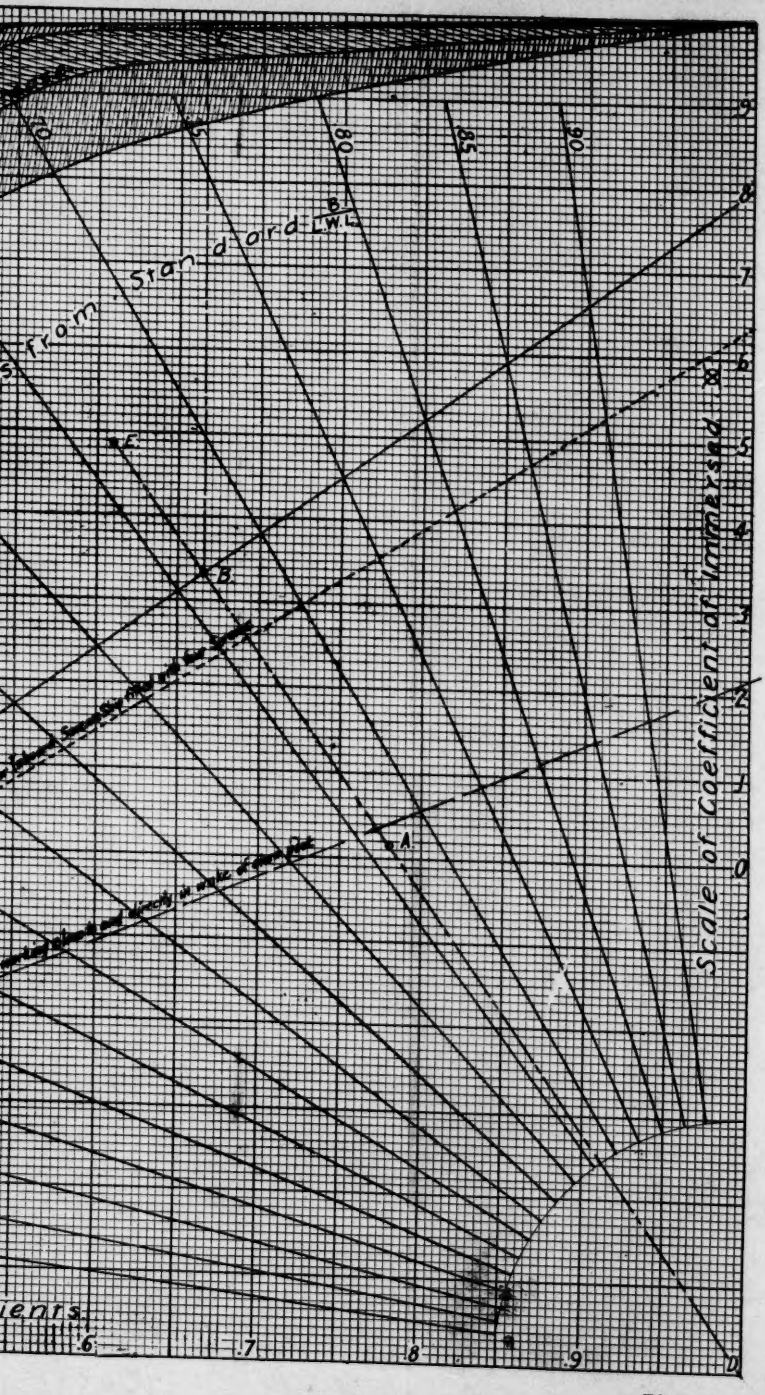
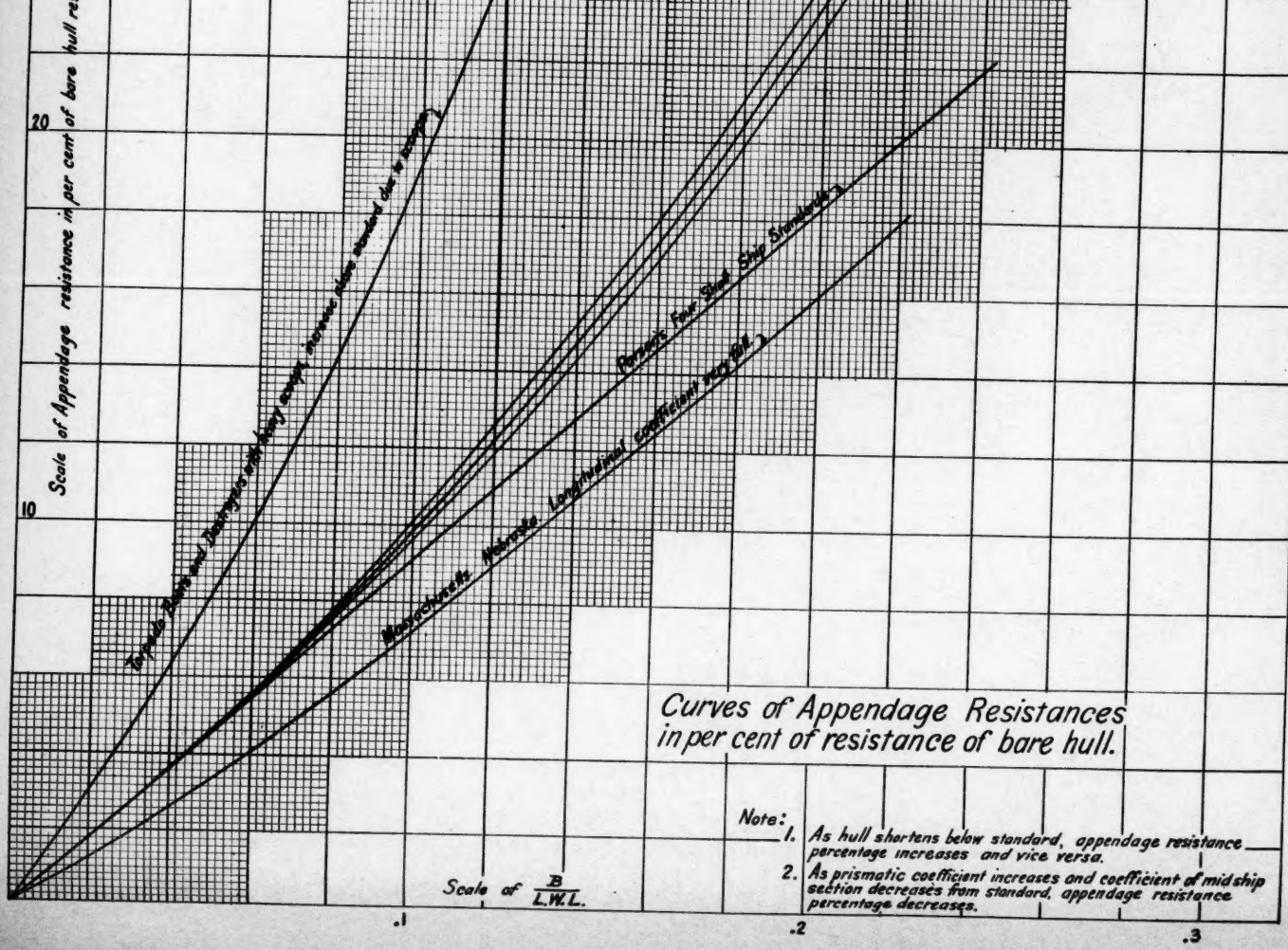

.80
.85
.90
from Stand-ard-ord-B/W.L.
Scale of Coefficient of Immersed
E.
B.
A.
.6
.7
.8
.9

Plate 1.

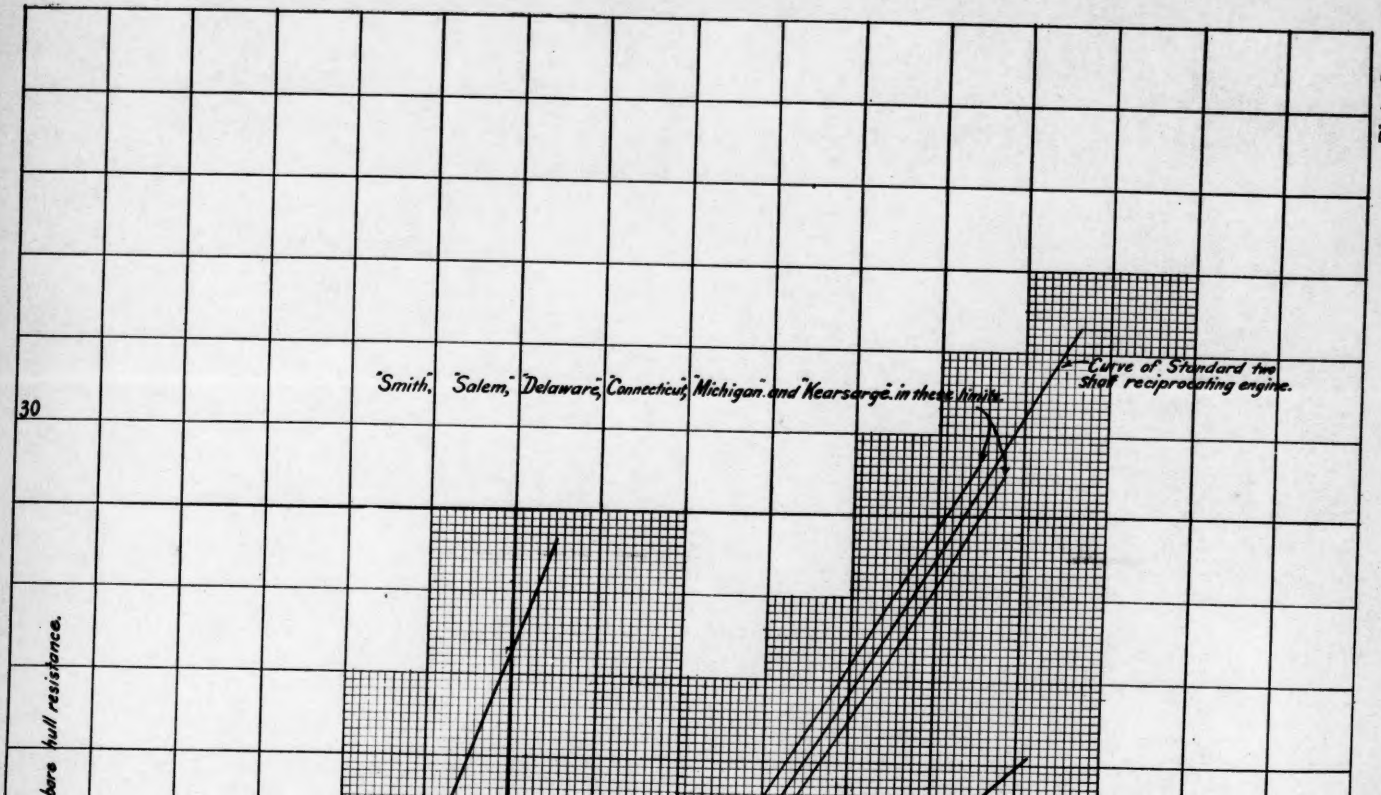

Scale of Appendage resistance in per cent of bare hull re
20
10
Torpedo Boats and Destroyers ordinary except screws also rotated due to struts
Battleships, Battle Cruisers, Longitudinal coefficient very full
Cruisers, Fine Lined Ship Standard
Scale of B/L.W.L.
.1
.2
.3

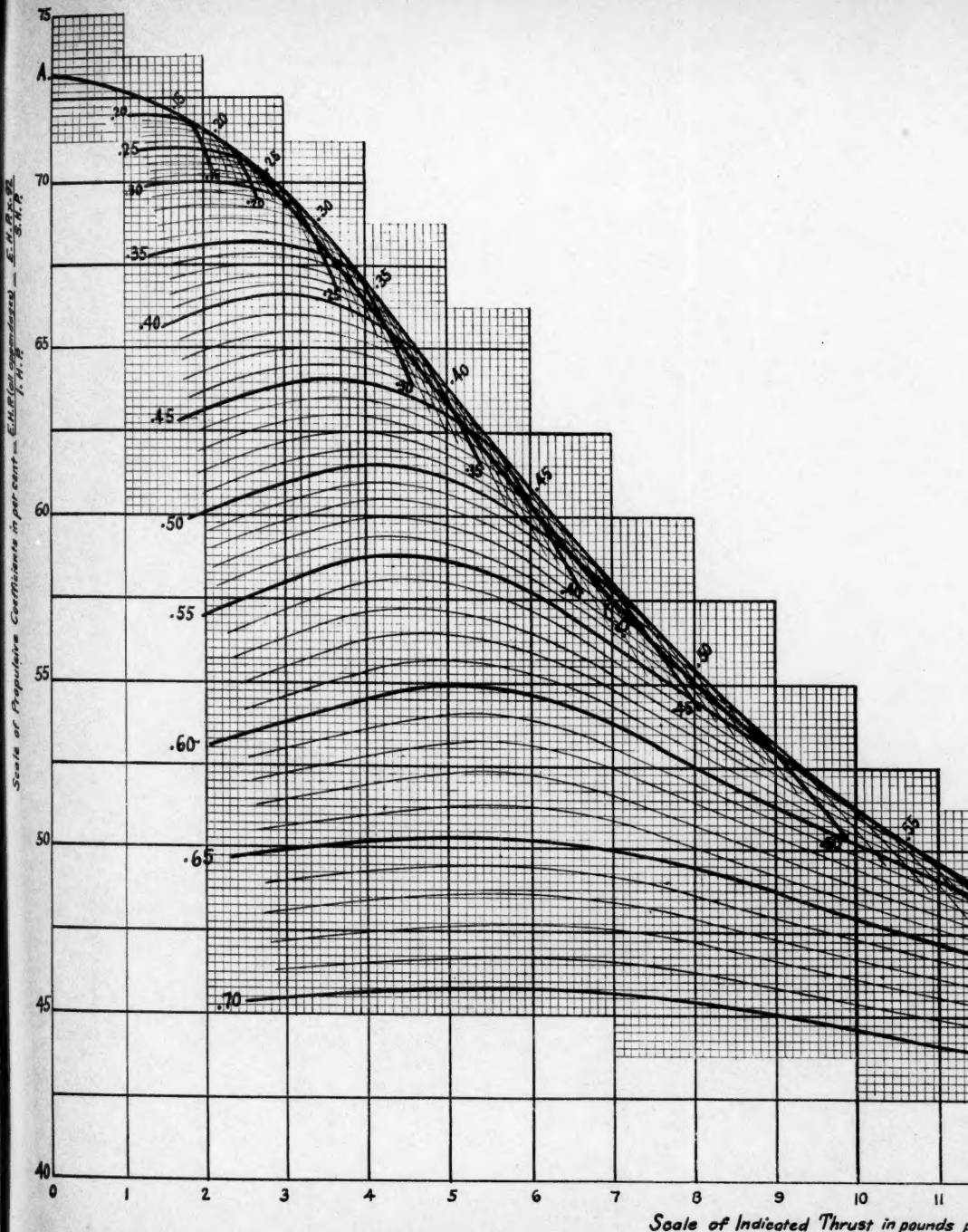
Curves of Appendage Resistances in per cent of resistance of bare hull.

Note:

1. As hull shortens below standard, appendage resistance percentage increases and vice versa.
2. As prismatic coefficient increases and coefficient of midship section decreases from standard, appendage resistance percentage decreases.

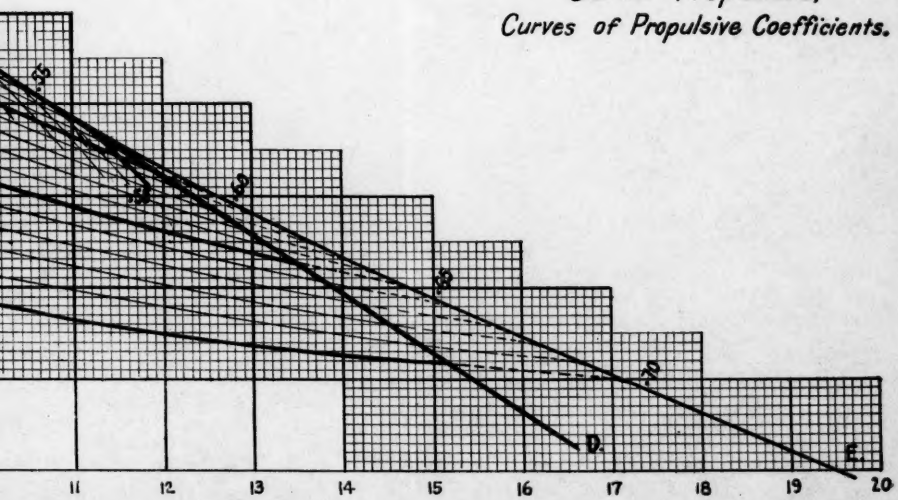
Plate 2.


30
"Smith," "Salem," "Delaware," "Connecticut," "Michigan" and "Kearsarge" in these limits.
Curve of Standard two shaft reciprocating engine.
bare hull resistance.


Scale of Propulsive Coefficients in per cent = E.H.P.(at propeller) / I.H.P. E.H.P.=T.V. S.H.P.
75
70
65
60
55
50
45
40
A.
.20
.25
.30
.35
.40
.45
.50
.55
.60
.65
.70
0
1
2
3
4
5
6
7
8
9
10
11
Scale of Indicated Thrust in pounds

Screw Propellers.

Curves of Propulsive Coefficients.



in pounds per sq. in. of Disc Area.

Plate 3.

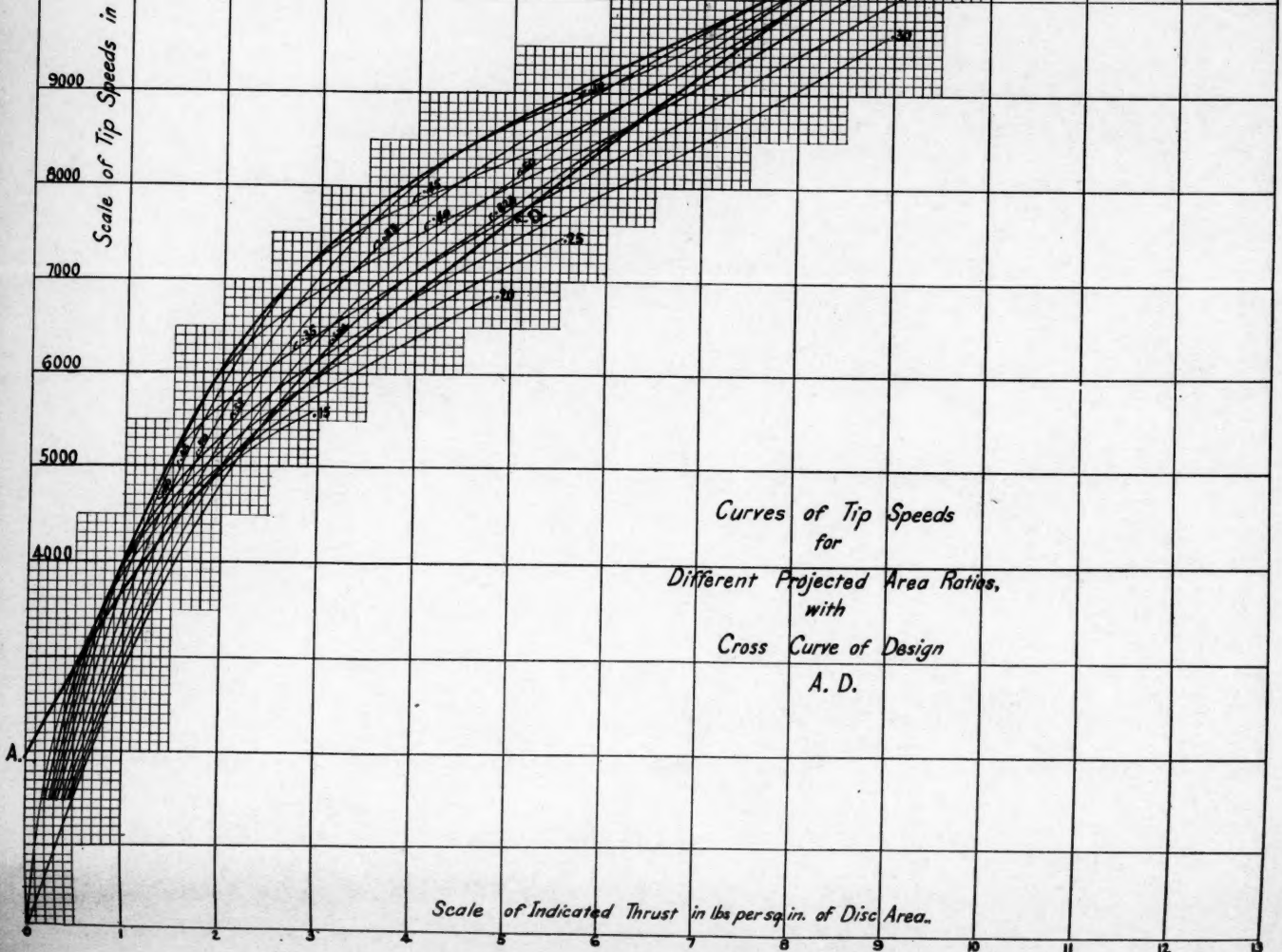
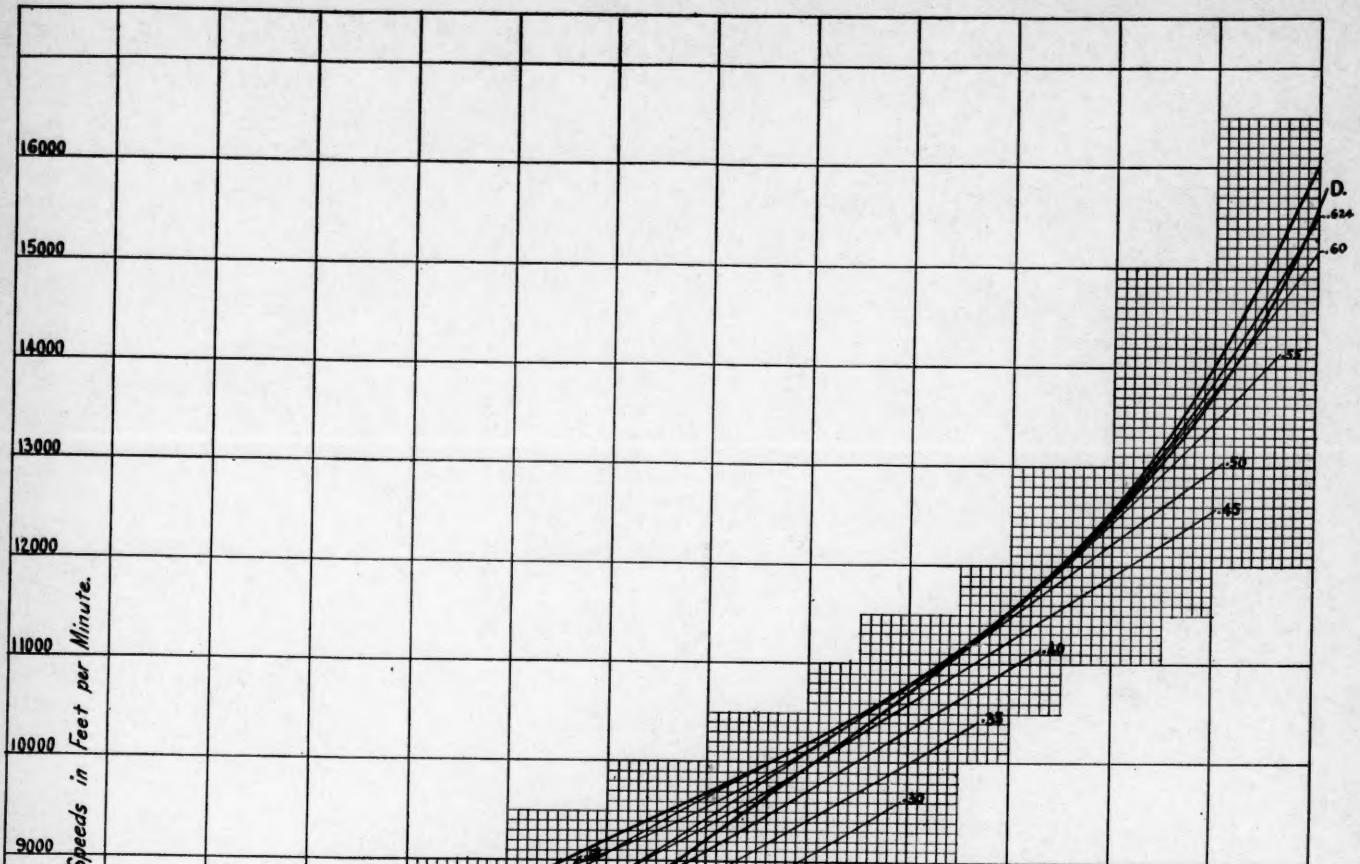

Scale of Tip Speeds in
9000
8000
7000
6000
5000
4000
A.
.30
.25
.20
.15
Curves of Tip Speeds
for
Different Projected Area Ratios,
with
Cross Curve of Design
A. D.
Scale of Indicated Thrust in lbs per sq in. of Disc Area.
0
1
2
3
4
5
6
7
8
9
10
11
12
13

Plate 4.

Plate 4.


16000
15000
14000
13000
12000
11000
10000
9000
Speeds in Feet per Minute.
D.
.624
.60
.55
.50
.45
.40
.35
.30

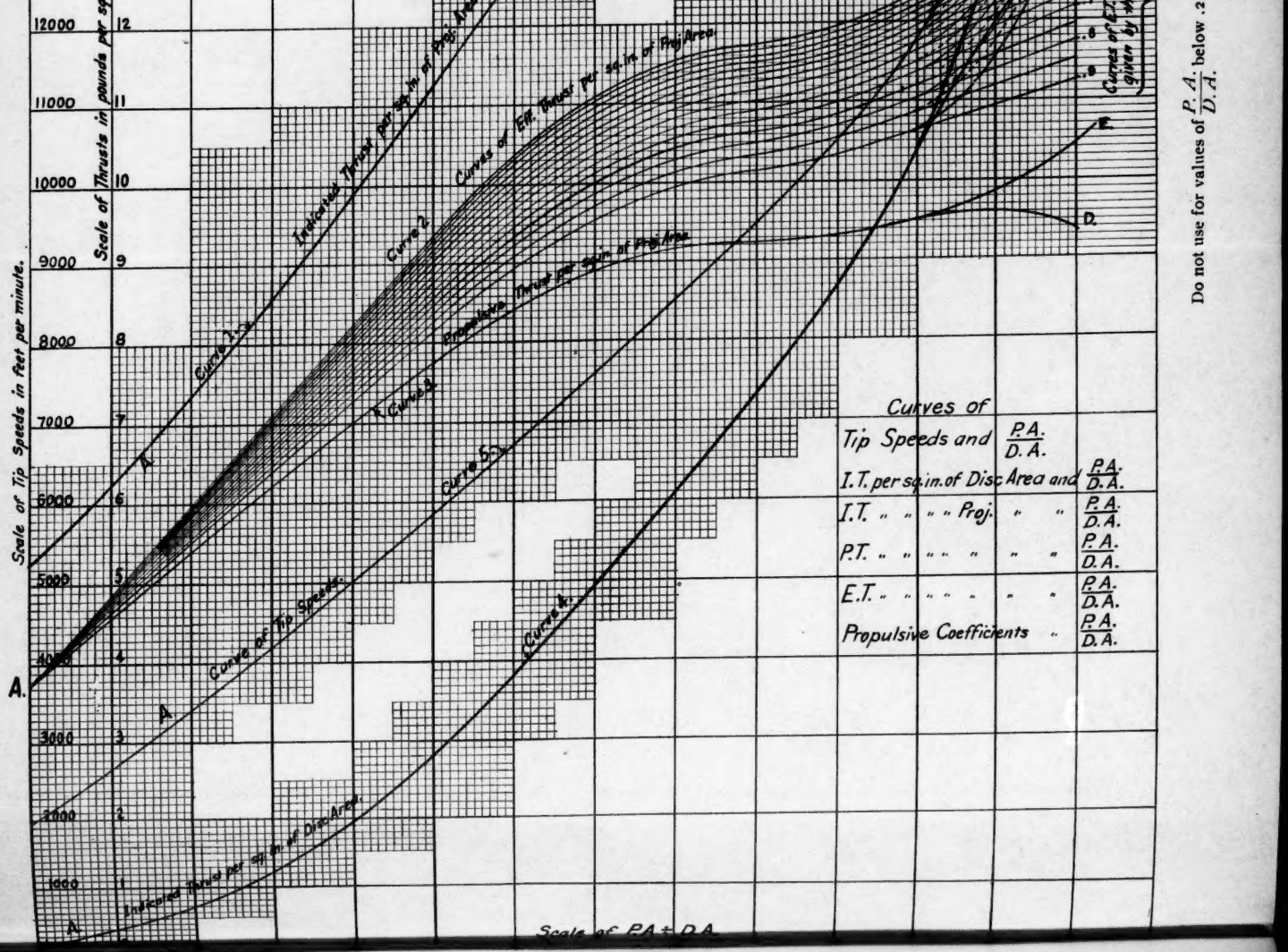
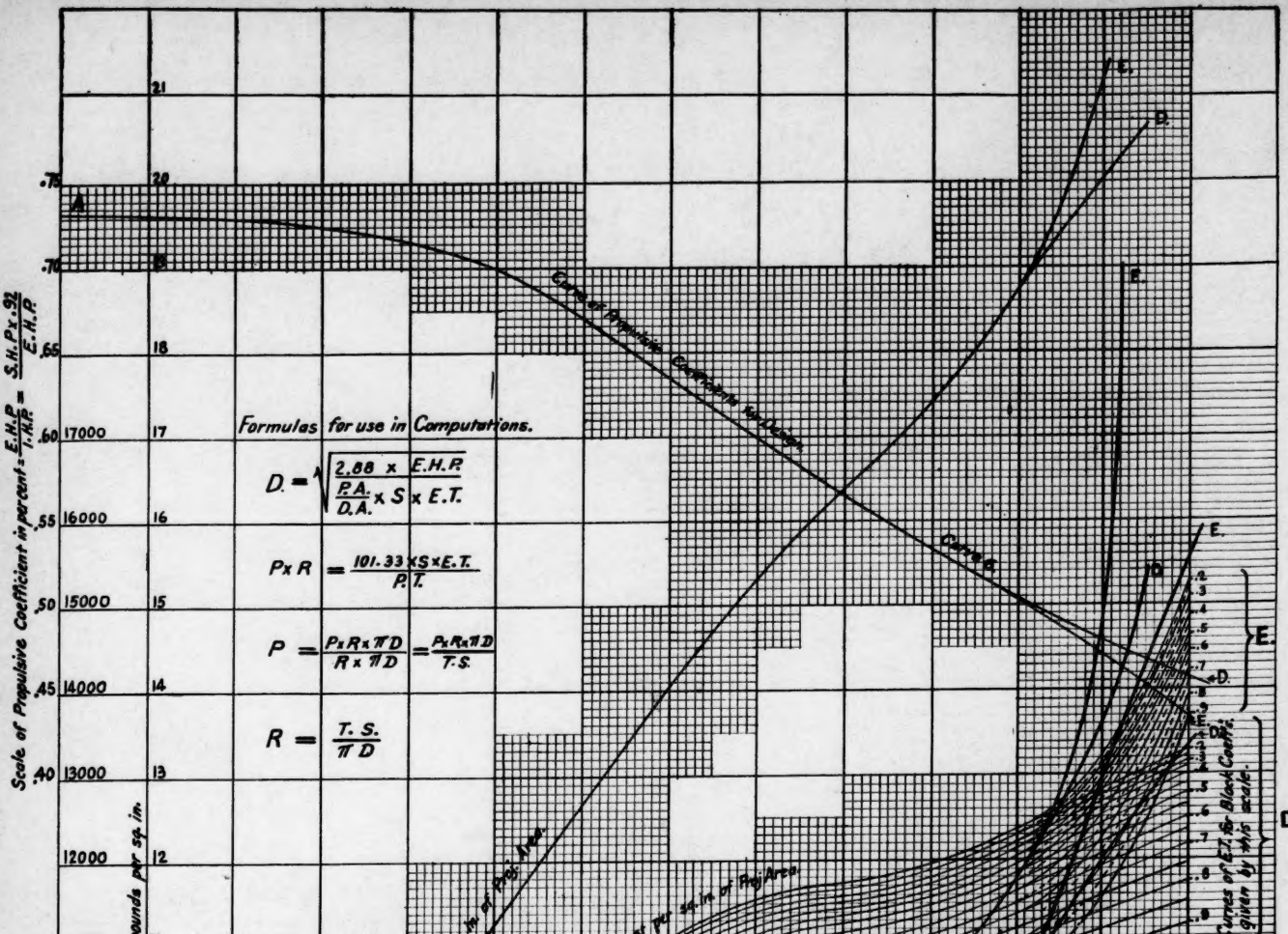

Scale of Tip Speeds in Feet per minute.
Scale of Thrusts in pounds per sq. in.
12000
11000
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
12
11
10
9
8
7
6
5
4
3
2
1
Indicated Thrust per sq. in. of Proj. Area
Curves of Eff. Thrust per sq. in. of Proj. Area
Curve 1
Curve 2
Propulsive Thrust per sq. in. of Proj. Area
Curve 3
Curve 5
Curve 4
Curve of Tip Speeds.
Indicated Thrust per sq. in. of Disc Area.
Curves of Tip Speeds and P.A./D.A.
I.T. per sq. in. of Disc Area and P.A./D.A.
I.T. " " " " Proj. " " P.A./D.A.
P.T. " " " " " " P.A./D.A.
E.T. " " " " " " P.A./D.A.
Propulsive Coefficients " P.A./D.A.
Scale of P.A. ÷ D.A.
Do not use for values of P.A./D.A. below .2

Plate 5.



Curves of $\frac{E.H.P.}{D.}$ below .2.

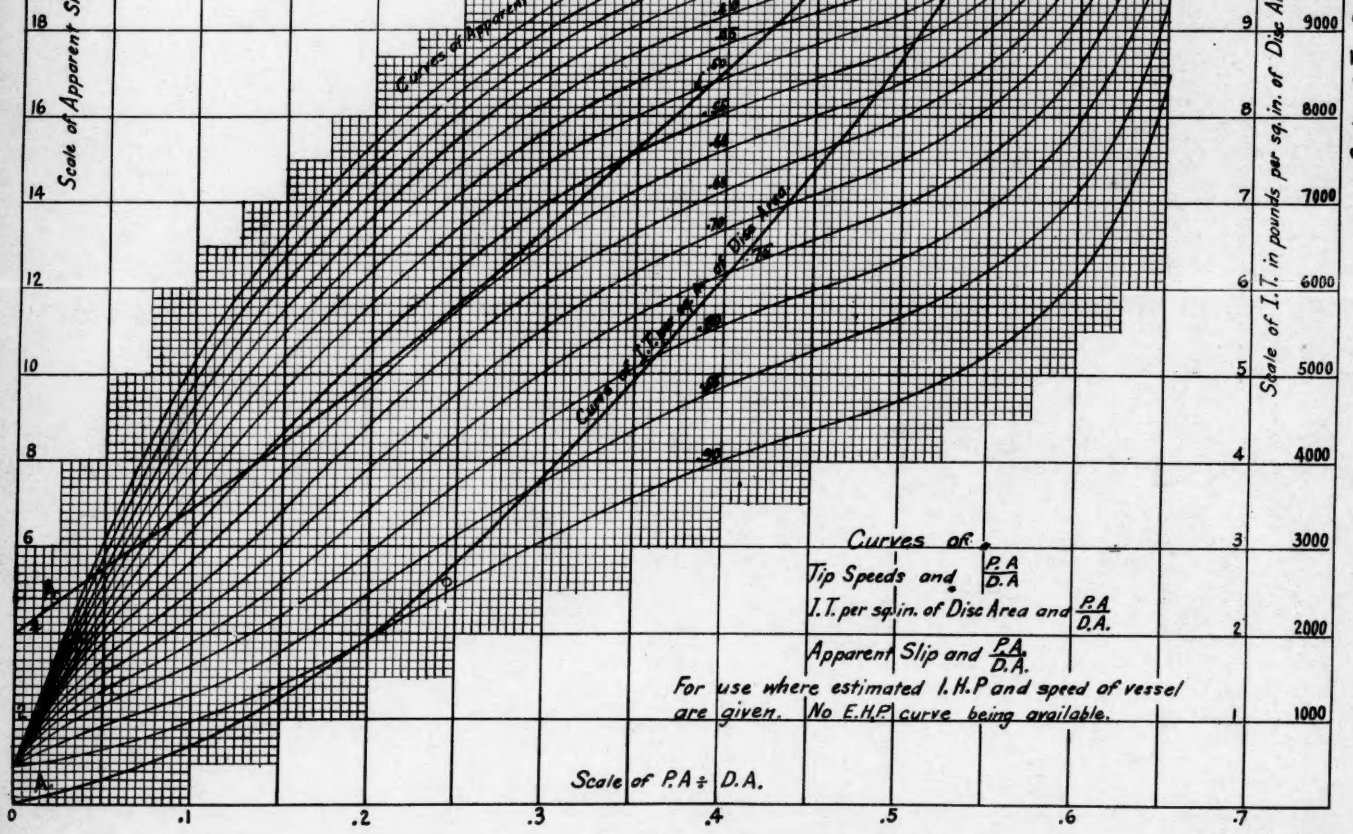
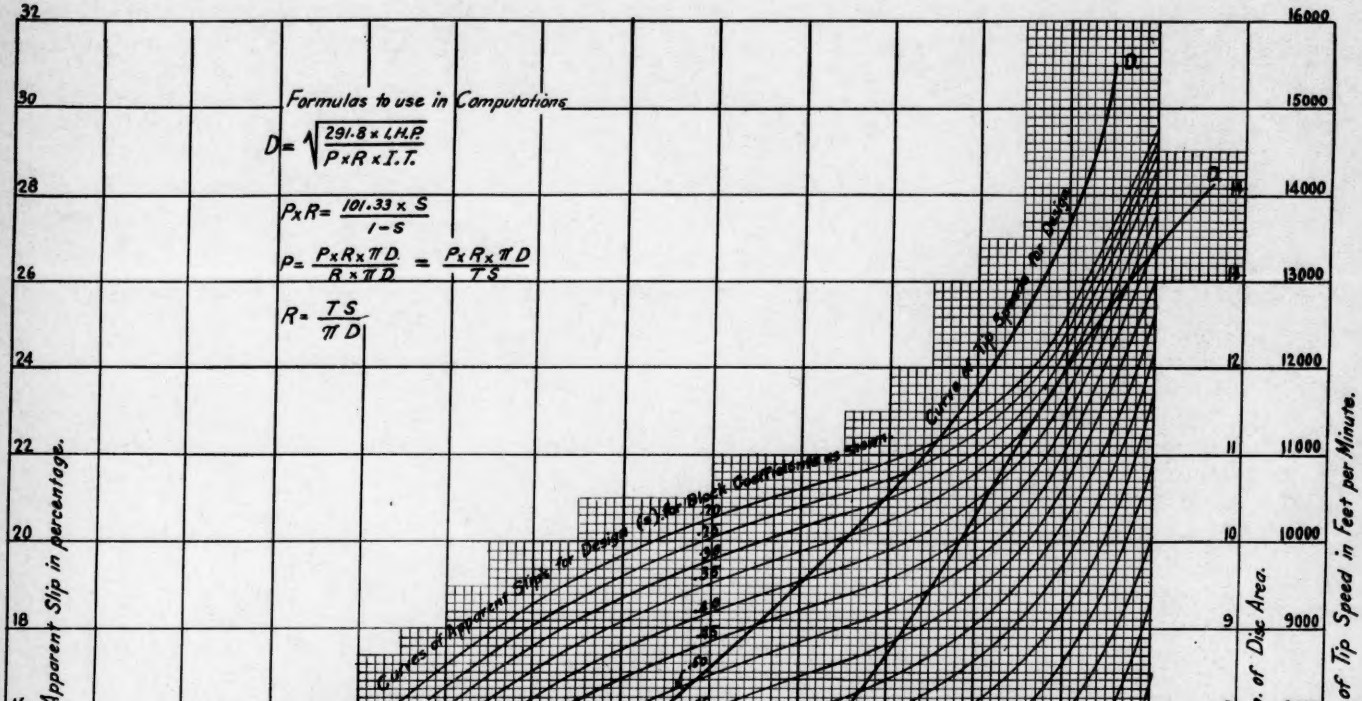
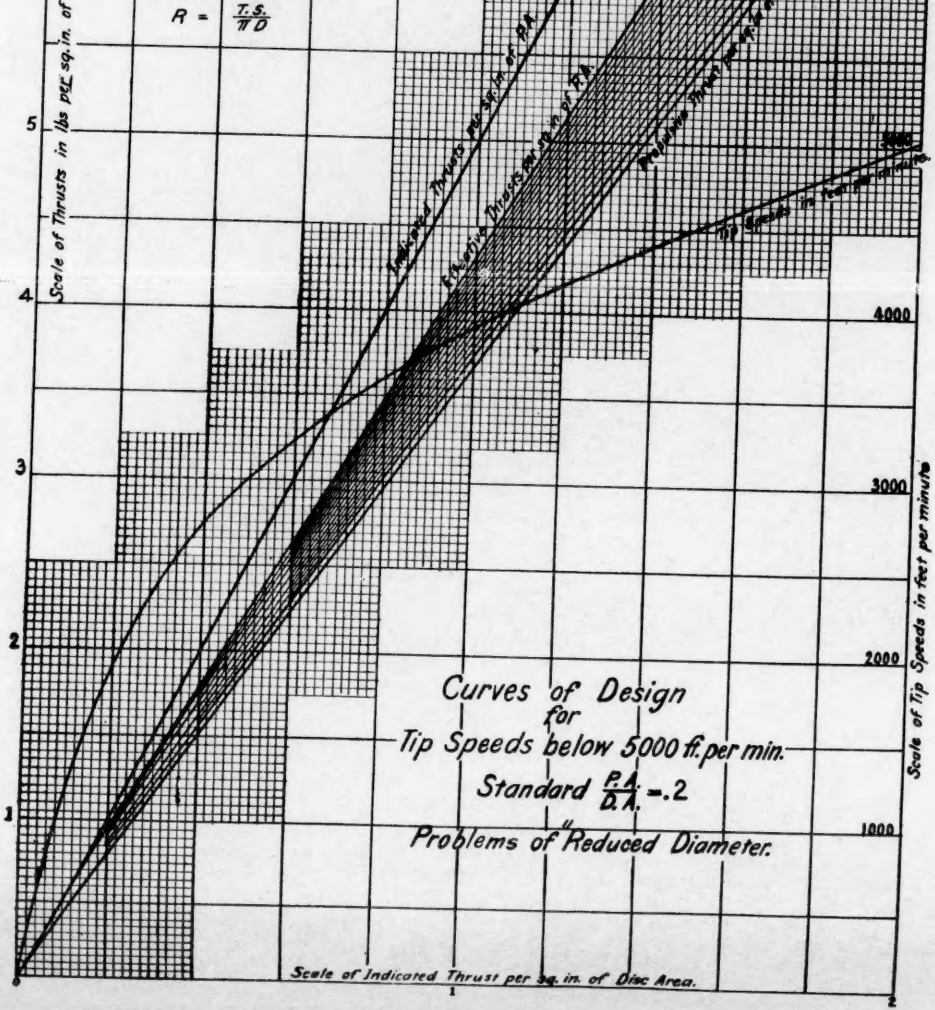

Scale of Apparent Slip
Scale of I.T. in pounds per sq.in. of Disc Area
Scale of Tip Speed
Curves of
Tip Speeds and P.A / D.A
I.T. per sq.in. of Disc Area and P.A / D.A.
Apparent Slip and P.A / D.A.
For use where estimated I.H.P and speed of vessel are given. No E.H.P. curve being available.
Scale of P.A ÷ D.A.
0
.1
.2
.3
.4
.5
.6
.7

Plate 6.

Do not use for values of $\frac{P.A.}{D.A.}$ below .2.


Formulas to use in Computations
$D = \sqrt{\frac{291.8 \times I.H.P.}{P \times R \times I.T.}}$
$P \times R = \frac{101.33 \times S}{1 - S}$
$P = \frac{P \times R \times \pi D}{R \times \pi D} = \frac{P \times R \times \pi D}{TS}$
$R = \frac{TS}{\pi D}$
Curves of Apparent Slip for Design (x) for Blade Coordinates as shown
Curve of Tip Speed for Above
% Apparent Slip in percentage.
Diam. of Disc Area.
Tip Speed in Feet per Minute.
32
30
28
26
24
22
20
18
16000
15000
14000
13000
12000
11000
10000
9000
D
D
12
11
10
9


$R = \frac{T.S.}{\pi D}$
Scale of Thrusts in lbs per sq. in. of
Indicated Thrust per sq. in. of P.A.
Effective Thrust per sq. in. of P.A.
Profile Thrust per sq. in. of P.A.
Speed in feet per minute
5
4
3
2
1
4000
3000
2000
1000
Scale of Tip Speeds in feet per minute
Curves of Design
for
Tip Speeds below 5000 ft. per min.
Standard $\frac{P.A.}{D.A.} = .2$
Problems of Reduced Diameter.
Scale of Indicated Thrust per sq. in. of Disc Area.
0
1
2

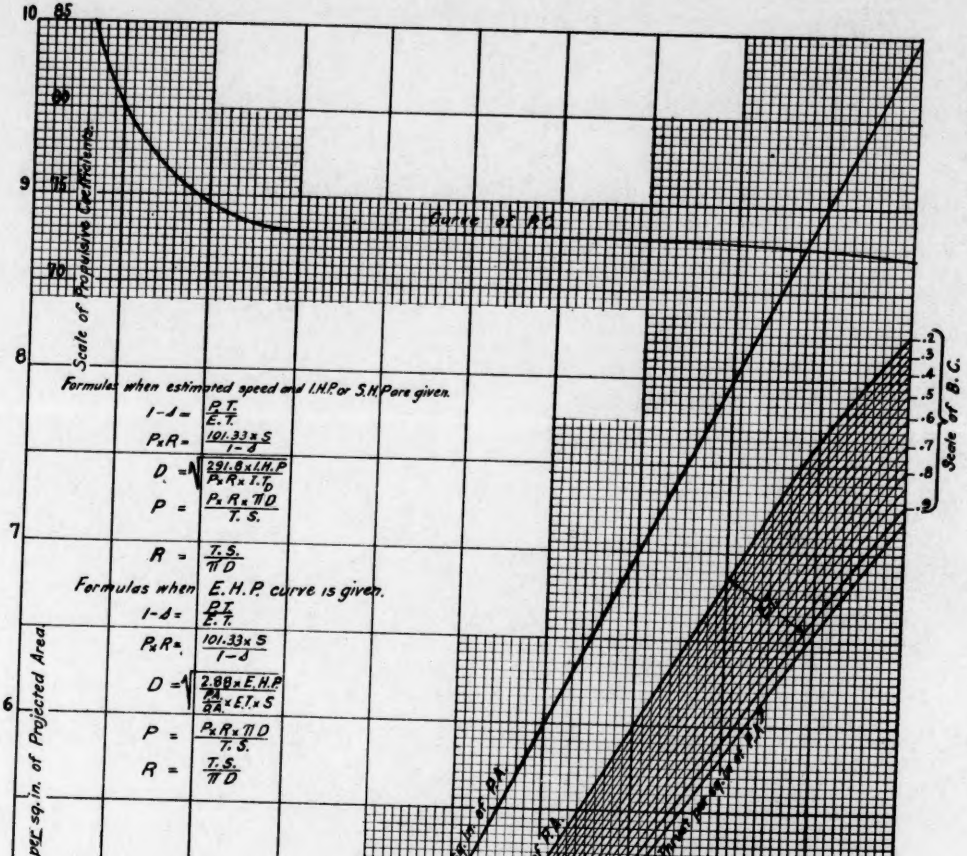
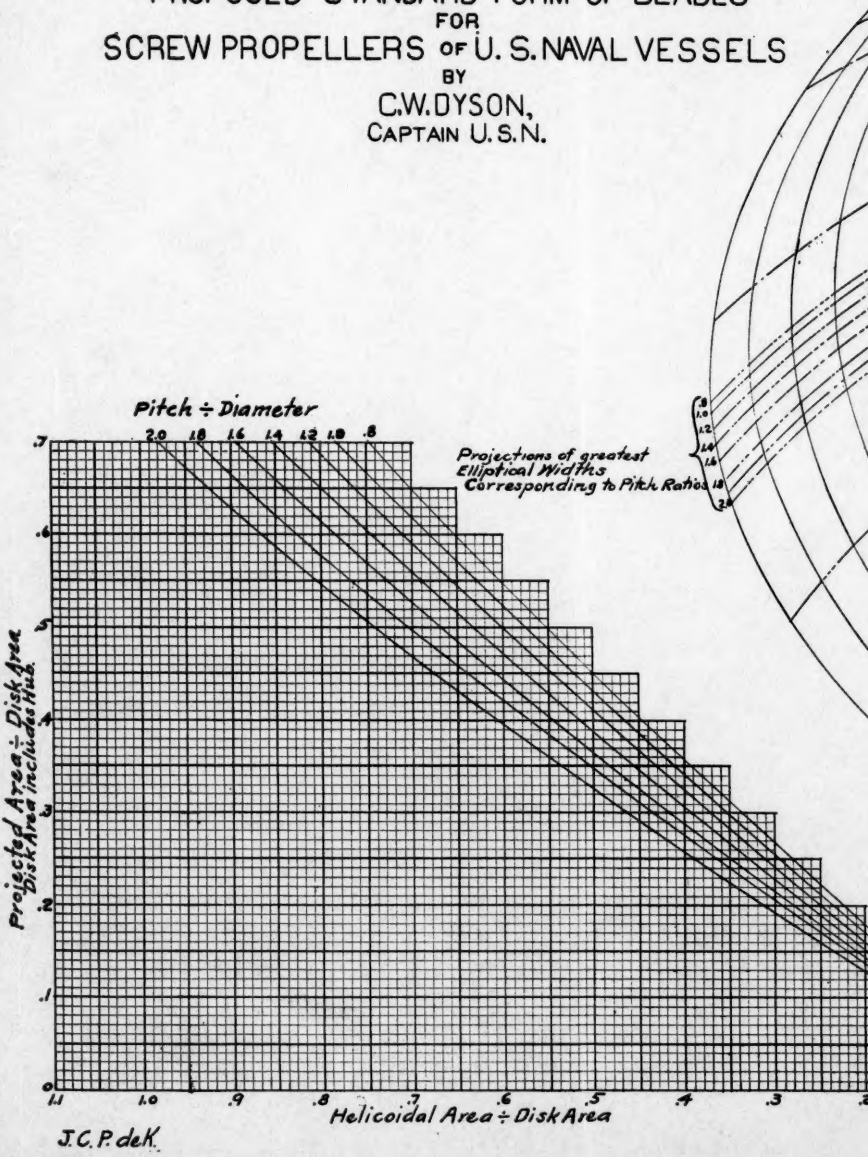

10
9
8
7
6
85
80
75
70
Scale of Propeller Coefficients.
Curve of R.C.
Formulas when estimated speed and I.H.P. or S.H.P. are given.
1 - Δ = P.T. / E.T.
P x R = 101.33 x S / 1 - Δ
D = √(291.8 x I.H.P / P x R x I.T.)
P = P x R x π D / T.S.
R = T.S. / π D
Formulas when E.H.P. curve is given.
1 - Δ = P.T. / E.T.
P x R = 101.33 x S / 1 - Δ
D = √(2.88 x E.H.P / PA/DA x E.T. x S)
P = P x R x π D / T.S.
R = T.S. / π D
Lbs. per sq. in. of Projected Area
Curve of PA/DA
Curve of PA/DA
Curve of PA/DA
.2
.3
.4
.5
.6
.7
.8
.9
Scale of B.C.

Plate 7.

PROPOSED STANDARD FORM OF BLADES
FOR
SCREW PROPELLERS OF U. S. NAVAL VESSELS

BY
C.W. DYSON,
CAPTAIN U.S.N.



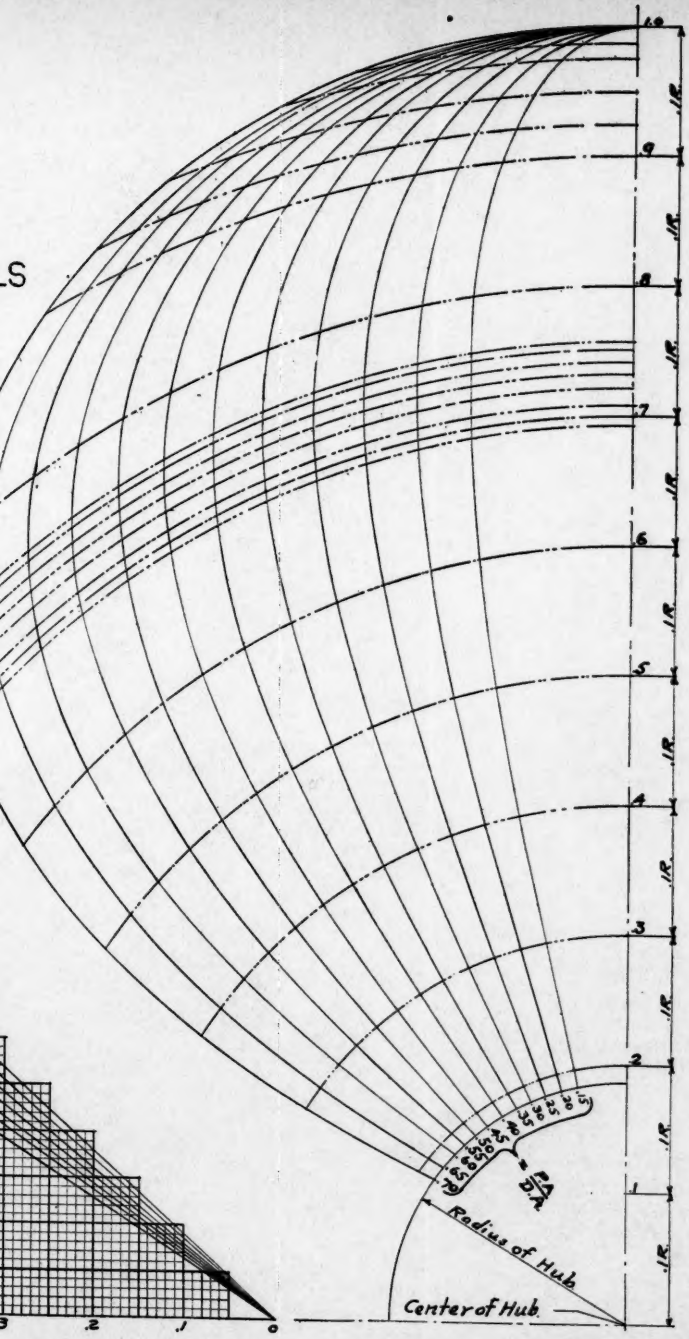

10
9
8
7
6
5
4
3
2
1
1R
15
20
25
30
35
40
45
50
55
60
P/A = D/A
Radius of Hub
Center of Hub
.3
.2
.1
0

Plate 8.

$\frac{P.A.}{D.A.}$	$\frac{E.T.}{\text{lbs.}}$	$\frac{P.T.}{\text{lbs.}}$	$\frac{P.T.}{E.T.}$	T. S.	E. H. P.	P. C.	H. P.	S	$101.33S$	$\frac{P \times R}{101.33S \times E.T.}$ / $P.T.$	D ft.	P ft.	R	Total = $\frac{S.H.P.}{.92\ H.P. \times 2}$
.20	8.09	7.03	.869	5,000	2,000	.715	2,797	28	2,837.24	2,365.0	11.28	33.13	141.1	5,137
.25	9.18	7.77	.8464	5,860	2,000	.70	2,857	28	2,837.24	3,352.1	9.468	17.01	197.0	5,257
.30	10.17	8.41	.827	6,700	2,000	.67	2,985	28	2,857.24	3,430.8	8.211	13.21	259.7	5,472
.35	10.88	8.90	.818	7,600	2,000	.634	3,155	28	2,857.24	3,468.6	7.35	10.54	329.1	5,805
.40	11.31	9.16	.8099	8,530	2,000	.599	3,339	28	2,837.24	3,504.2	6.743	8.79	402.6	6,144
.45	11.51	9.22	.801	9,500	2,000	.565	3,540	28	2,837.24	3,542.2	6.302	7.382	479.8	6,513
.50	11.71	9.32	.796	10,520	2,000	.5325	3,756	28	2,837.24	3,564.4	5.927	6.31	565	6,911
.55	12.15	9.51	.7827	11,800	2,000	.504	3,968	28	2,837.24	3,625.0	5.548	5.355	677	7,301
.60	12.61	9.60	.7613	13,600	2,000	.479	4,175	28	2,837.24	3,726.8	5.214	4.489	830.2	7,682
.625	12.86	9.55	.7426	15,300	2,000	.4675	4,278	28	2,837.24	3,820.7	5.059	3.969	962.6	7,872

Finally, to show how the propeller characteristics vary with the value of $\frac{P.A.}{D.A.}$ for any given block coefficient of hull, *E. H. P.* and speed, let the following be taken as an example:

Hull, standard form; block coefficient, .35; speed, 28 knots; *E. H. P.* for this speed = 4,000. Find *P*, *D*, *R* and *H. P.* for a series of screws varying from .2 to .625 *P. A.* ratio to answer these conditions; power on two screws: (page 627.)

While the method of design described in the foregoing text does not provide for exceeding the limit *A. D.* and is therefore not so elastic as the method described in preceding papers, it is much safer. With the method described previously, where the curves of apparent slip for each projected-area ratio were given, there was a constant temptation to obtain increased revolutions by holding to the lower projected-area ratios and using higher thrusts, as the lower *P. A.* ratios give higher propulsive efficiency at reduced powers. This led into danger, as with the low projected-area ratios the danger limit is reached rapidly with a very slight increase in indicated thrust over the safe thrust to use.

With the present method, if the lower projected-area ratios do not give the revolutions desired, it becomes necessary to advance into the higher ratios or to solve by the method of "reduced diameters."

NOTES.

PROPULSIVE MACHINERY AND OIL FUEL IN THE U. S. NAVAL SERVICE.*

By CAPTAIN C. W. DYSON, U. S. NAVY.

GENERAL INTRODUCTION.

Until within the last few years the improvements in propelling machinery for naval vessels, and for marine purposes in general, were few, the designers apparently considering that the reciprocating engines which they were then using were good enough, and that any further improvements in them could only be made at an undesirable increase in weight, in cost, and in complication.

This apparent view extended not only to the main propelling machinery but also to the auxiliary machinery, with the result that each new ship was practically a copy of those that preceded it, only such modifications being made as the necessities of the particular case dictated.

With the advent of the turbine it became necessary, in case the reciprocating engine was to hold its own, to take advantage of every possible opening for improvement. Such improvements as appeared possible at the time of laying down the designs were made, and that they were desirable has been amply shown by the results on trial and in service obtained by the U. S. S. *South Carolina, Michigan* and *Delaware*.

In the adoption of turbine machinery, the Navy Department proceeded with characteristic caution, and no designs of this type of machinery were laid down until results obtained abroad were of such nature as to practically insure success.

In the fall of 1904, it was determined to lay down the three scout cruisers, *Birmingham, Salem* and *Chester*, and, in order to obtain data for use in future designs, it was decided to fit the *Birmingham* with reciprocating engines, the *Salem* with Curtis turbines and the *Chester* with Parsons turbines.

Before these vessels were completed, in June, 1907, the Fore River Shipbuilding Company completed and tried out the Southern Pacific Steamer *Creole*, which was fitted with Curtis turbines as main propelling engines. The trials were witnessed by representatives of the Navy Department who, entirely discounting the fact that such a vessel as the *Creole* was unfitted by her hull and slow speed for turbine propulsion, reported on the turbines only, and after pointing out defects which could be corrected, stated that the Curtis turbine was adapted for marine propulsion.

During this same year, 1907, the designs for the battleships *Delaware* and *North Dakota* were being prepared. When the advertisement for bids for these two vessels were issued, bids were invited both for reciprocating engines and for turbines. The design of these vessels made it impossible to install satisfactory Parsons turbines, but single-unit Curtis

* Read before the Engineers' Club of Philadelphia, January 6, 1912.

turbines fitted in very well. Upon the opening of the bids, the Bureau of Steam Engineering pointed out to the Navy Department that if turbine machinery were installed in either of these vessels it would be at the sacrifice of cruising radius. How well based this criticism was has since been amply demonstrated by the performances of the *Delaware* with reciprocating engines, the *North Dakota* with Curtis turbines, and the *Utah* with Parsons turbines.

PROPELLING MACHINERY OF FAST AND LIGHT VESSELS.

For main propelling engines for such vessels as destroyers and torpedo boats, the steam turbine was welcomed with open arms. For such vessels having high speed, high power and very light foundations for the support of the propelling machinery, a very high piston speed was necessary with the reciprocating engines which had formerly been fitted. This high piston speed, with the consequently high number of reversals in the direction of motion of the moving parts, resulted in the production of excessive vibrations of hull and machinery, and the resultant breakdowns of the engines at full power were numerous.

By the adoption of turbines, with the consequent change from reciprocating to rotary motion, the problem of engine vibration was immediately solved, and the only vibration now existing in such vessels is that due to the propellers.

The first turbine-propelled destroyers, Nos. 17 to 21, five in number, were laid down in 1906, and all the destroyers designed, built and building, since that year, a total of 29, have been fitted with turbines. Some of these vessels have the Parsons turbine, with the power distributed between three shafts, while others have the Curtis or the Zoelly, with the power distributed on two shafts.

IMPROVEMENTS IN RECIPROCATING ENGINES.

For battleship work, in order to meet the turbines on more even grounds, the principal changes made in the design of reciprocating engines are as follows:

1. Increase in ratio of L.P. to H.P. cylinder volumes from about 7 to 1 to about 10 to 1.
2. Lengthening the main steam valves in order to give short and straight steam and exhaust ports with consequent reduction in clearances and in steam frictional losses.
3. Increase in vacuum carried in main condensers.
4. Use of superheated steam.
5. Slight decrease in bearing pressures of crank pins and crossheads.
6. The fitting of forced lubrication to crank-shaft, crank-pin and crosshead journals and to eccentrics and to crosshead slides.

All of these changes have not as yet been made, but 1, 2 and 4 were utilized in the engines of the *South Carolina* and *Michigan*, 1, 2, 4, 5 and 6 in the engines of the *Delaware*, and 1, 2, 3, 5 and 6 in the engines of battleships Nos. 36 and 37, the designs of which have just been completed by the Navy Department.

RESULTS OBTAINED BY THE ABOVE CHANGES.

Unfortunately, in the cases of the *South Carolina* and *Michigan*, no measurements of actual water consumptions of the machinery were made on the trials of these vessels. An idea of the economy realized can be obtained, however, by comparing them with their sister ships of the *Connecticut* class, the *Delaware* also being included in the tabulation:

.....	*Minnesota.*	*Michigan.*	*Delaware.*
Heating surface of boilers, square feet.	52,752	42,500	55,749
Superheating surface of boilers, sq. ft.	...	4,720	6,149
Total grate surface per sq. ft., equiv...	...	886	1,106
Total grate surface per sq. ft., actual..	1,100	1,048.25	1,439
I.H.P. all machinery.....	20,783	19,680.6	29,529
Sq. ft. of H. S. per I.H.P.....	2.538	2.159	1.888
Sq. ft. of H. S. per I.H.P. (total).....	...	2.4	2.096
I.H.P. per sq. ft. grate surface, equiv.	...	22.213	26.7
I.H.P. per sq. ft. grate surface, actual.	18.894	...	...
Air pressure, in inches of water, equiv.	...	1.006	2.43
Air pressure, in inches of water, actual.	.92	.85	1.86

Let us compare the above results still further, using a series of vessels having the same type boilers as the above vessels, but all being without superheat and having the old type of naval engines.

In the above table and in the following one the boilers are all reduced to a standard ratio of generating heating surface to grate surface of 48.1 and the air pressures calculated as varying inversely as the ratio of the equivalent grates to the actual ones.

VESSELS WITH OLD TYPE ENGINES AND NO SUPERHEAT.

.....	*Minnesota.*	*Montana.*	*Mississippi.*	*Charleston.*
Heating surface.....	52,752	68,000	32,648	64,000
Total grate surface, actual...	1,100	1,690	768	1,400
Total grate surface, equiv....	...	1,418	681	1,334
I.H.P. all machinery..	20,783	28,280	13,906	27,507
Sq. ft. of H. S. per I.H.P.	2.538	2.404	2.347	2.323
I.H.P. per sq. ft. G. S., act..	18.894	...	...	...
I.H.P. per sq. ft. G. S., eq..	...	20.0	20.405	20.62
Air press., ins. of water, act.	.92	1.19	1.3	2.48
Air press., ins. of water, eq..	...	1.33	1.47	2.6

Plotting the results obtained from the above tables on Plate I, we see that for an air pressure of 2 inches of water, which is the naval limiting pressure for coal-burning Babcock & Wilcox boilers, which are fitted in all the above vessels, the use of superheat and the improved engines gives 25.6 I.H.P. per square foot of grate against 20.6 I.H.P. per square foot of grate with the old equipment. This is a total increase in economy of 24.3 per cent. The commonly accepted saving due to superheat is estimated at 1 per cent. for every ten degrees, which, as to average superheat used in the above vessels at full power, amounted to about 60 degrees, gives as a total saving due to superheat only, of 6 per cent., leaving a balance of 18.3 per cent. due to the improvements in the engines themselves.

COMPARISON OF WATER RATES OF OLD AND NEW TYPES.

The only vessels having reciprocating engines for which accurate engine water consumption measurements have been taken up to the present date are the scout *Birmingham* and the battleship *Delaware*. The necessary engine data for comparision is as follows:

.....	*Birmingham.*	*Delaware.*
H.P. cylinder, inches	28½	38½
Diameters I.P. cylinder, inches	45	57
2 L.P. cylinders, inches	62	76
Piston-rod diameters, inches	6	8
Stroke of piston, inches	36	48
Revolutions at full power	191.5	128.39
Piston speed at full power, per minute, feet	1,149	1,027.12
Per cent. clearances, H.P., mean	26.4	16.17
I.P., mean	21.34	13.57
L.P.	21.245	12.49
Steam pressure per gage in H.P. valve chest	229.4	253

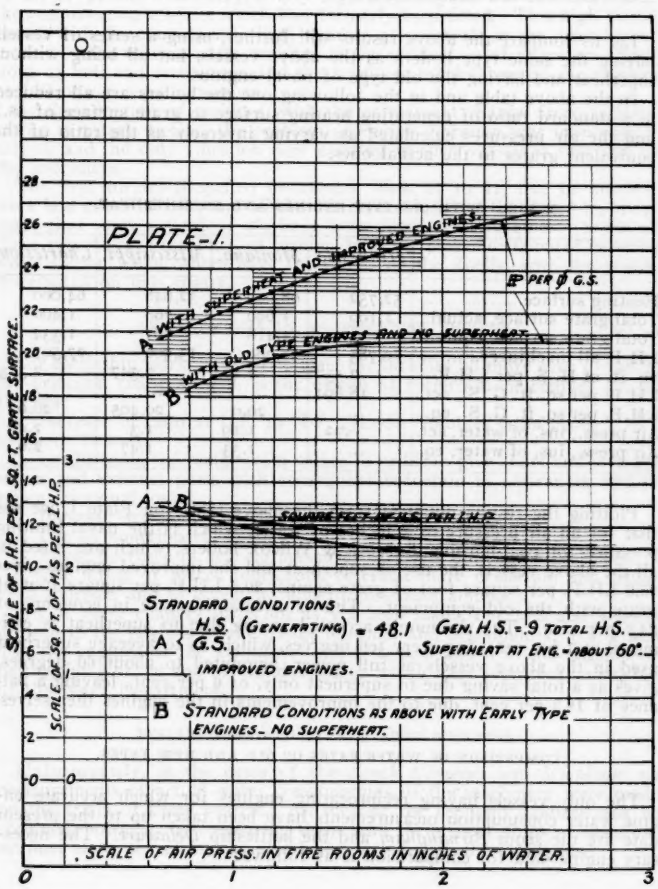
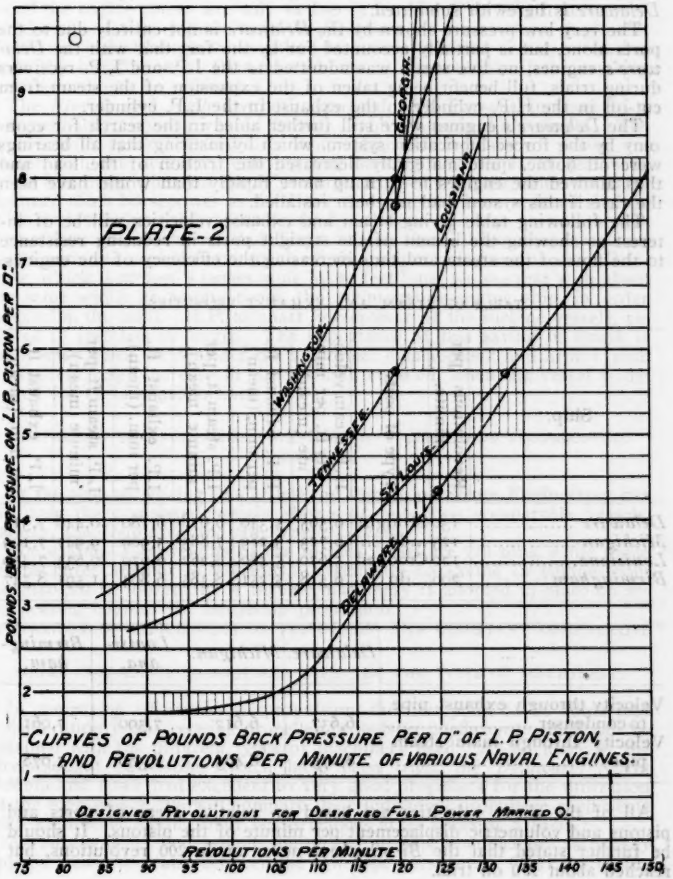


PLATE-I.
A-WITH SUPERHEAT AND IMPROVED ENGINES.
B-WITH OLD TYPE ENGINES AND NO SUPERHEAT.
A-B
SQUARE FEET OF H.S. PER I.H.P.
SCALE OF I.H.P. PER SQ. FT. GRATE SURFACE.
SCALE OF SQ. FT. OF H.S. PER I.H.P.
STANDARD CONDITIONS
A H.S. (GENERATING) / G.S. = 48.1 GEN. H.S. = .9 TOTAL H.S.
SUPERHEAT AT ENG. = ABOUT 60°.
IMPROVED ENGINES.
B STANDARD CONDITIONS AS ABOVE WITH EARLY TYPE ENGINES - NO SUPERHEAT.
SCALE OF AIR PRESS. IN FIRE ROOMS IN INCHES OF WATER.

The *Birmingham* engine, of course, is handicapped at the start by its smaller size, higher piston speed and lower steam pressure, but these handicaps cannot possibly explain away the great differences existing between the steam consumptions per I.H.P. of the two types of engines at equal fractions of power. These consumptions are as follows:

.....	Full power.	$\frac{9}{16}$ power.	$\frac{1}{2}$ power.
Delaware	13.38 lbs.	12.7 lbs.	15.12 lbs.
Birmingham	17.3 lbs.	15.5 lbs.	19.00 lbs.
Per cent decrease for *Delaware*	22.66	18.7	20.4



The above amounts include drainage from the jackets. All leakage from stuffing boxes is unaccounted for, but if this were taken into account the difference would be still more favorable to the *Delaware*, as not the slightest evidence of leakage through her valve-stem or piston-rod stuffing boxes existed during the trials.

The collier *Cyclops* has engines built along the same lines as the modern naval reciprocating engines, and the results obtained by her bear out fully the results obtained by the *Delaware*. Although no superheat exists in the case of the *Cyclops*, the water per I.H.P. measured from her indicator cards, shows at full power less than 12 pounds, neglecting leakage and cylinder condensation.

To show the effect of the short straight steam ports on exhaust pressures, Plate 2 showing the amount of back pressure exerted per square inch of L.P. piston area for several engines of the old type and of the *Delaware* is herewith submitted.

The very low pressure shown by the *Delaware* is not entirely due to the ports alone, but is partially accounted for by the fact that with the *Delaware's* engines, no live steam was admitted to the I.P. and L.P. receivers during trials, full benefit being taken of the expansion of the steam from cut-off in the H.P. cylinder to the exhaust in the L.P. cylinder.

The *Delaware's* engines were still further aided in the search for economy by the forced-lubrication system, which by insuring that all bearings were oil borne, quite materially decreased the friction of the load and thus allowed the engines to turn up more rapidly than would have been the case if this system had not been installed.

The following table giving steam and exhaust velocities will be of interest as showing the benefit of the straight ports in reducing resistance to the flow of the steam and thus increasing the efficiency of the engines.

TABLE OF STEAM AND EXHAUST VELOCITIES.

Ship.	Revolutions per minute.	Type of port.	H.P. steam velocity ft. per minute (mean).	H.P. exhaust ft. per min. (mean).	I.P. steam ft. per minute (mean).	I.P. exhaust ft. per min. (mean).	L.P. steam ft. per minute (mean).	L.P. exhaust ft. per min. (mean).
Delaware	130	Straight	6,565	5,340	6,678	6,180	10,446	7,514
Michigan	125	...do	6,723	5,289	7,883	5,909	9,947	7,199
Louisiana	120	Crooked	5,670	5,480	8,262	6,670	10,833	7,450
Birmingham	200	...do	6,518	5,251	8,385	6,804	11,591	8,347

.....	*Delaware.*	*Michigan.*	*Louisiana.*	*Birmingham.*
Velocity through exhaust pipe to condenser	6,637	6,612	7,390	7,061
Velocity through main steam pipe	8,275	8,463	8,380	7,078

All of the above velocities are based on relative areas of ports and pistons and volumetric displacement per minute of the pistons. It should be further stated that the *Birmingham* never made 200 revolutions, but reached about 190 on trial.

TURBINE MACHINERY FOR BATTLESHIPS.

In the choice of type of turbine machinery for battleship propulsion the Navy Department has, up to the present time, placed turbines of the Parsons type and of the Curtis type upon an equal footing where weights and space allowed for the installation would accommodate either. In some cases the engine-room space may be very much restricted; when this is the condition, turbines of the Curtis type have a decided advantage over the Parsons, as the former type with either one or two units per shaft, the total power being divided between two shafts, can be installed in much smaller floor area than the Parsons type with its cruising turbines and high-pressure backing turbines, in addition to the main turbines, with the power distributed among four shafts.

An instance of this occurred in the case of the *North Dakota.* In this vessel the engine rooms are only 44 feet in length. This length accommodated reciprocating engines, as in the *Delaware,* or a nine-stage single-unit Curtis turbine, although this resulted in a very crowded arrangement, but made the installation of Parsons turbines an impossibility if the cruising turbines were to be retained.

The *Florida* and *Utah* are both equipped with Parsons turbines of a designed shaft horsepower of 28,000, divided among four shafts. The engine-room space required to accommodate this amount of power, together with the necessary cruising and backing turbines is 51 feet wide by 60 feet long, while on the *Wyoming* and the *Arkansas,* with the same designed shaft horsepower as the *Utah,* the space has been reduced slightly to 48 feet 6 inches wide by 60 feet. In all four of these vessels the engine rooms are much crowded, large as they are. When they are compared with the engine-room space of the reciprocating-engine ship *Delaware,* which is 50 feet 6 inches wide by 44 feet long, we see that with about the same width of engine room, the reciprocating-engine ship, in order to develop the same I.H.P. as shaft horsepower of the turbine vessels, requires 16 feet less in length. The importance of this saving in length is easily seen when we consider the additional displacement and armor, with consequent large increase in cost of hull, entailed, when the vessel is designed for turbine propulsion.

COMPARISON OF ENGINE-ROOM WEIGHTS OF TURBINE AND RECIPROCATING ENGINES.

At the time of writing this article the Bureau of Steam Engineering has available the completed weight sheets of only two comparable vessels, namely, the *Delaware* and the *North Dakota.*

The engine-room weights of the former vessel are 773 tons, while those of the latter amount to 783 tons. Practically no saving in engine-room weights can be looked for if turbine machinery is adopted in place of reciprocating engines for battleship propulsion.

COMPARISON OF EFFICIENCIES OF PROPULSION AND RESULTANT COMPARATIVE BOILER INSTALLATIONS REQUIRED WITH THE TWO TYPES OF MACHINERY.

In comparing efficiencies of propulsion, the first point to be taken into consideration is the relative propeller propulsive coefficients which can be obtained with the different types of machinery used. As all three of the Dreadnought battleships which have been tried, the *Delaware, North Dakota* and *Utah,* had excellent to very good propellers for the utilization of the high power, we will compare their individual propulsive coefficients with each other, taking their performances at 21 knots as a basis.

As all engineers understand, the term "propulsive cofficient" as usually applied means the ratio between the tow-rope horsepower (E.H.P.) of the

ship when fitted with all appendages and the I.H.P. in the case of reciprocating engines, or S.H.P. in the case of turbines. The Bureau of Steam Engineering assumes a mechanical efficiency of 92 per cent. for its large reciprocating engines, so that the true comparison of propulsive coefficient would be represented by the following formulas:

$$\text{Reciprocating propulsive coefficient} = \frac{\text{E.H.P.}}{\text{I.H.P.}};$$

$$\text{Turbine propulsive coefficient} = \frac{\text{E.H.P.} \times .92}{\text{S.H.P.}}.$$

Using these two formulas, the propulsive coefficients of the three vessels at 21 knots were:

Delaware.....122½ revolutions, 65 per cent. propulsive coefficient.
North Dakota.....265 " 53.82 per cent. propulsive coefficient.
Utah.....313 " 56.12 per cent. propulsive coefficient.

While the powers required for this speed were:

Delaware.....23,400 I.H.P.
North Dakota.....26,500 S.H.P. = 28,800 I.H.P.
Utah.....26,400 S.H.P. = 28,700 I.H.P.

For boiler comparisons it is considered preferable to reduce conditions to one common condition of feed temperature and boiler pressure, and to compare the heat units absorbed by the boilers of each vessel in overcoming equal resistances of hulls. By doing this we find that at 12 knots speed the *Delaware's* boilers absorbed 119,500 B.T.U.'s against 142,700 B.T.U.'s for the *Utah*, or that it costs approximately 19.4 per cent. more to cruise at 12 knots with the turbine ship than with the reciprocating one. At full power of 21.56 knots the costs in heat units for the two vessels are approximately the same, namely, about 385,000 B.T.U.'s.

The above heat costs include all the engineer's and the ship's auxiliaries. The *North Dakota* is not brought into this comparison, as in the measurements of water consumption on that vessel only the engineer's auxiliaries were included.

This equality of heat cost at full power existing for battleships of 21 knots speed, it is not readily seen how a saving of 15 per cent. in boiler weights, as has been claimed, can be made by the adoption of turbines for main propelling machinery in such cases.

CRUISING ECONOMY.

As pointed out above, the *Utah* at 12 knots required the absorption of 19.4 per cent. more heat units by her boilers than did the *Delaware*. Referring to equal speeds and comparing the *Utah's* performance with those of the *Delaware* and the *North Dakota*, we obtain the results given below:

Ship.	Speed.	S.H.P.	I.H.P.	Revs.	Per cent.
Delaware	12	...	3,800	66¼	69.3
North Dakota	12	3,750	4,076	140¼	61.32
Utah	12	3,800	4,130	172	53.18

The above results are all based upon acceptance-trial results and would be considerably modified in actual service, as was shown by the performances of the *Delaware* and *North Dakota* when cruising in company with the Fleet.

The reciprocating-engine ship in bad weather showed up as fully 20 per cent. better than the turbine ship, and in good weather at a slightly higher speed, as nearly 44 per cent. better. These results were obtained on the cruise of the Fleet to England and back home. In justice to the turbine it must be stated that the turbines of the *North Dakota*, when examined shortly after, were found to be in damaged condition due to erosion and corrosion, although the vessel was a new ship just starting a cruise.

POINTS OF SUPERIORITY CLAIMED FOR TURBINE MACHINERY OVER RECIPROCATING MACHINERY FOR BATTLESHIP PROPULSION.

These are:

1. Capable of being driven for long periods of time at high powers without the liability to derangement which exists with reciprocating engines.
2. Less work required to keep them in condition, and therefore decreased engine-room force.
3. Less vibration, and therefore resultant better gun pointing.
4. Decreased boiler power necessary to develop power required for full speed.
5. Less total floor space necessary for engines and boilers.
6. Greater economy in oil and fuel at full power.
7. Greater economy in oil at cruising speeds.
8. Greater cleanliness of engine room.
9. Greater capacity for overload.
10. Greater ease of repair.
11. Maintenance of original economy due to no increase of leakages from wear.

Let us take these claims, item by item, and see whether they are justified by actual experience.

1. There is no doubt about the ability of the turbine to stand up for long periods of time when driven at full power, but cannot the reciprocating engine when properly designed do the same thing? Experience with the *Delaware* would seem to demonstrate that it can.

Immediately upon the return of this vessel from Chili and just before she arrived at Boston, she received a wireless message from the Navy Department directing that she proceed to sea as soon as possible after coaling ship and carry out speed trials as follows:

Four hours at the maximum speed possible, to be immediately followed, without any interval for overhaul, by a twenty-hour run at the highest speed that could be maintained for that length of time.

In carrying out these orders the *Delaware* proceeded to Boston and began coaling as soon as possible. At the end of about 20 hours, during which time not a bearing nor cylinder was opened for adjustment or inspection, she proceeded to sea, and as soon as well clear of the land began the trials.

During the first four hours the average speed maintained was 21.86 knots per hour, while for the entire 24 hours the average speed was about 21.3 knots per hour, the decreased speed for the 24 hours over that for the first four-hour period being due to fires and boilers becoming dirty and the fireroom personnel fatigued, thus automatically slowing down the ship and easing up the work on the engines as the time became more extended. It is unnecessary to point out that this same automatic easing up would have occurred with any type of steam-driven machinery where coal was used as the fuel.

Upon the completion of the trial the commanding officer of the vessel reported to the Department practically as follows: "Four hours and 20 hours speed trials completed. Not the slightest disarrangement of machinery, either main engines or auxiliaries. Ship ready for immediate duty."

(2) (10). These claims may be discussed as follows:

The turbine by its very nature is less liable to slight derangements than the reciprocating engine, but when derangements do occur they are usually very serious and require the vessel to proceed to a dock yard for repair, where, as the troubles are usually blading ones, the time for repair stretches into weeks, thus losing the services of an important unit from the fighting line for the entire time of repair. With the reciprocating engine, however, such repairs as are usually called for are within the capacity of the ship's force, and the ship can remain on her station while the repairs are being made; in other words, turbine sickness usually requires hospital treatment while the reciprocating engine can be cured by home treatment. Furthermore, there is as much difference between the two types when they are out of order as there is between an animal and a human being. Both give indications of being under the weather, but the reciprocating engine tells you where the trouble exists, while with the turbine you know that something is wrong inside, but what is wrong is known only after the hospital surgeons get to work.

As to the decreased engine-room force, our turbine battleships carry and need just as large a force as do the reciprocating ships of the same class.

(3) As to vibration, where the reciprocating engine is well balanced and is mounted on a heavy, substantial hull, such as that of a battleship, the vibration is barely noticeable, and in the case of the *Delaware* and *North Dakota* there appeared to be fully as much vibration caused by the propellers with the turbine vessel as there existed with the *Delaware*. Furthermore, if gun pointing be taken as a measure of vibration, it is pointed out that the *Delaware* has just won the battle-practice trophy, beating out her turbine-propelled sister quite badly.

(4) As for claim No. 4 it has been shown that with turbines for the designed full speed of the vessel a considerable increase in power over that required with reciprocating engines is made necessary by the decreased propulsive efficiency of the propellers. When turbine propulsion first became an accomplished fact, in all reports of trials great stress was laid upon the fact that the water consumption per hour per S.H.P. of the turbines was very considerably less than per I.H.P. with reciprocating engines, but the great difference between the propulsive efficiencies of the propellers was not mentioned, this great inferiority of the turbine propeller for battleship work practically requiring the same amount of boiler power to be fitted for turbines as for reciprocating engines.

(5) This claim is not justified, as Curtis turbine installations require approximately the same space, and Parsons turbines with cruising turbines more floor space, than is required for a reciprocating-engine installation for the same speed of vessels.

(6), (7), (8). Claim 6 is correct so far as lubricating oil is concerned, for with forced lubrication of reciprocating engines and with an open-top oil casing, oil is thrown on the lower cylinder heads where it vaporizes and is lost. The saving-of-fuel claim does not exist. Claim 8 is correct, but only at high powers. To offset these claims, the working platforms of reciprocating-engine installations are much more habitable on account of lower temperature.

(9) This claim is justified only by the fact that turbines are not designed so close to the power requirements as are reciprocating engines. The steam areas through the blading are much larger than are necessary for the passage of the steam required for the designed power at the designed pressure. This criticism applies more particularly to turbines of the Parsons type.

(11) This claim is a fact so far as the claim for no increase in leakage of steam is considered, but it can hardly be called an advantage over the reciprocating engine, as with proper care and experience the valves and pistons of the latter machine can be kept as tight throughout their service

as they are they day they are put in commission. To offset this turbine claim we have the following troubles to encounter: excessive corrosion of rotors, which destroys balance; unequal expansion of rotors and of rotors and casing, which destroys clearance and results in destruction of blading; clearance troubles due to wearing of turbine and thrust bearings and the necessity of micrometer adjustments of these bearings to maintain clearances.

VESSELS OF HIGH POWER AND SPEED, SUCH AS BATTLESHIP CRUISERS, SCOUTS AND DESTROYERS.

When powering such vessels as the above the power required for the high speed becomes very great, and in order to confine ourselves to units of light weight it is necessary to divide the power between two, three or four shafts. Should reciprocating engines be used, they would be designed for a high number of revolutions and high piston speeds, and would thus become much more liable to derangement than are the comparatively slow-running massive engines of the ordinary *Dreadnought* battleships. In such cases we turn to the turbine as the rational power producer, and up to date for such purposes the turbine is the machine par-excellence.

In our service the battleship cruiser does not yet exist, our extra fast vessels at present consisting of three scouts and 34 destroyers built and building.

As already stated, one of the scouts is fitted with reciprocating engines. She has done very good service, and below 21 knots has been considerably more economical than her sister ships. Above that speed the turbine ships have the advantage, the percentage advantage increasing, as the speed increases above 21 knots. As with these types of vessels high speed is the prime requisite, and economy at cruising speeds, while desirable, but still only a secondary consideration, no hesitation is met with in discarding the reciprocating engine as the primary propelling machine and taking up the turbine in its place.

With the Parsons turbine in these fast vessels the power is usually divided between three or four shafts, additional turbines for cruising at low and moderate speeds being fitted. These cruising turbines have been the cause of the major part of our troubles with the Parsons type, probably 90 per cent. of the blade strippings which have occurred have happened in these turbines, and have usually taken place when the injured rotor was running idly in a vacuum. In the latest destroyers, in order to further increase the cruising economy and to escape the blade troubles of the cruising turbines, these last-named turbines have been omitted, and small rapid-running reciprocating engines installed in place of them. These engines are designed for about 350 revolutions and are fitted with forced lubrication, the working parts being enclosed in an oil casing. They remain in operation, exhausting through the H.P. turbine until the speed reaches about 16 knots. Above this speed they will be disconnected and the main turbines only will be used, the H.P. turbine being fitted with a couple of cruising stages for use in producing speeds from 16 to about 25 knots.

This cruising reciprocating engine has been also fitted, one engine on each shaft, to two-shaft arrangements, where the Curtis or the Zoelly turbines are used as main propelling engines. With such an arrangement, however, the cruising engines are an additional weight over the already quite heavy weights of these impulse-turbine units, and the advantages gained will have to be considerable to justify their retention.

The main point of difference in the different designs of these combination systems is in the designed exhaust pressure from the L.P. cylinder of the reciprocating engines to the inlet nozzle of the turbine, and the

stage of the turbine at which this exhaust steam is admitted. It may be accepted as an axiom that "the higher the exhaust pressure from the engine to the turbine the greater will be the range of speeds through which the combination will hold its superiority over the straight reciprocating or the straight turbine drive."

In the Fore River design this exhaust pressure is taken at about five pounds absolute, and is admitted into the fourth stage of the turbine, while in the Bureau designs the exhaust pressure from the reciprocating engine is taken at about 25 pounds absolute and is admitted into the second main stage of the main H.P. turbine. With a low exhaust pressure from the engine to the turbine at full power of the reciprocating engine, the value of the turbine falls off very rapidly as the power of the reciprocating engine is reduced, and within a very short range of power the lower stages of the turbine will commence to do negative work, and thus produce a brake effect on the system resulting in actual loss.

THE CHOICE BETWEEN TWO AND THREE-SHAFT ARRANGEMENT FOR DESTROYERS.

The Navy Department, until the giving out of the contracts for the last destroyers designed, had made no distinction in favor of one arrangement over the other, but had laid down in the contract plans of the machinery that arrangement and type of turbine for which the greatest floor area of engine room was required. This was the Parsons type with three shafts. In inviting bids, however, bids were asked for on contractors' plans, in addition to those on the Department's plans, and bidders were always given to understand that in the consideration of bids one set of plans would receive as much consideration as the other.

The two-shaft arrangement does have the following advantages over the three-shaft arrangement, namely: Better maneuvering qualities at low speeds, better backing power, fewer propelling units and possibly a better propulsive efficiency of propellers. This latter point is not definitely settled, however, but the writer is of the opinion that with properly designed propellers this superiority of the two-shaft arrangement must exist, at the low speeds in particular.

When the last eight vessels were contracted for the Department seized the opportunity offered it by the various bids to obtain the above advantages, and, out of the eight, six boats of the two-shaft arrangement were taken, the remaining two being the three-shaft Parsons-turbine arrangement as shown on the Department's plans. Since closing the contract for these two revised plans have been approved, changing them to the two-shaft arrangement but retaining the Parsons turbines.

ECONOMY AND RUGGEDNESS OF THE DIFFERENT TYPES OF TURBINES.

Remarks under this head can be summed up in a very few words, as follows: Considerable trouble has been experienced with Parsons turbines due to stripping of blades through loss of clearance, while, since an initial fault in the L.P. blading of the Zoelly turbines of the *Warrington* and *Mayrant* has been corrected, blading troubles with the Curtis and the Zoelly turbines are practically unknown. These latter classes, due to their greater blade clearances, do not require the accurate adjustment that is absolutely necessary with the Parsons turbine, and from this fact is reaped a decided benefit in service. As to the relative economy of propulsion of the three types, the question is involved by the use of various designs of boilers with the different types. Results appear to indicate, however, that there is little difference between them, with the balance slightly in favor of the impulse-reaction type.

Leaving the subject of main propelling engines, we will now describe the most important departure from early practice that has been made in late years; this is the adoption of oil as a fuel.

OIL-FUEL SYSTEM OF THE NAVY.

In adopting oil fuel for the Naval Service the first thing necessary was to decide upon that system of atomization of the fuel which was best adapted to our needs and conditions. The system to be adopted must be one which would entail no loss in fresh water and the minimum additional weight possible.

The very conditions of the problem forced the Department into the search for a satisfactory method of mechanical atomization. By adopting such a method no loss of fresh water would occur and no air compressors were required. The only additional weights required with such a system were those of the necessary oil pumps, piping and burners.

After investigating the field available, the method of mechanical atomization as developed by the Schutte-Koerting Company was decided upon as the most promising for a foundation on which to build, and this system was adopted.

DESCRIPTION OF THE SYSTEM.

The system as developed for use in the Naval Service really could be classed as "Oil-Fuel Burning Reduced to the Simplest Form." It consists of oil-fuel supply or booster pumps, oil-fuel pressure pumps, oil piping and strainers, oil heaters, air cones and oil burners. In addition are the forced-draft blowers.

FUNCTIONS OF THE DIFFERENT PARTS.

Oil-Fuel Supply or Booster Pumps.—Handle the oil in the storage tanks, or bunkers, pumping from one tank to another, filling tanks from oilers alongside, draining tanks of settled water, aiding suctions of fuel-oil service pumps on long suction lines. They may be either simplex or duplex pumps.

Oil-Fuel Service Pumps.—Draw from the oil-fuel mains through strainers and discharge through strainers and through oil heaters to oil burners. These pumps are usually duplex, of long stroke, and are large enough to be slow running. They work at a discharge pressure not exceeding 225 pounds, this pressure being determined by the amount of oil discharged and the area of the discharge orifice in the burner nipple.

Oil-Fuel Heaters.—The function of these heaters is primarily to heat the oil to such a temperature as will render it sufficiently fluid to ensure efficient atomization. Secondarily, the added heat may possibly add slightly to the efficiency of combustion. The first idea in heating the oil was that it must be raised to a temperature slightly above the flash point, so that immediately upon issuing from the burner tip the oil assumes a gaseous form and bursts into flame. This high heating of the oil has been found to be unnecessary and is also objectionable as it produces carbonizing in the burners and pipes. It is also dangerous, as leaks in the piping lines cannot be discovered except by moving a naked flame along the line to ignite escaping gas. The temperature we now maintain is approximately 175 degrees F., any increase above that being of doubtful value.

Air Cones.—These are one of the most vital items in the entire oil-burning system. They are in the form of a truncated cone, and carry the burner in the axis of the cone, discharging towards the base. The air for combustion is admitted through slots with guide vanes around the cir-

cumference of the cone, and is given a whirling motion. The velocity and amount of air admitted may be regulated in two ways, either by a conical damper around the cone, as in the New York Shipbuilding Company installations, or by an air register carried directly across it, as in the Peabody system installed by the Cramp Ship and Engine Building Company. Both methods give excellent results.

The problem of air admission is of the greatest importance, as if it is improperly done smokeless combustion becomes nearly an impossibility, and the burning of oil at a high rate of combustion becomes accompanied by a series of pulsations of such magnitude as to cause the boiler casings to pant and the brick linings to break loose and fall.

Oil Burners.—These are of the simplest character and consist of an oil pipe with a cast piece screwed on the end and forming the tip. The opening in the tip varies from 1.5 m.m. to about 2.3 m.m., depending upon the maximum amount of oil to be burned. Inside of this tip casting is a whirling chamber to which the oil is admitted in such a manner as to give it a rapid whirling motion around the axis of the burner. This causes the oil issuing through the tip opening to fly off tangentially as soon as free from the burner, and produces an intimate mixture of the oil with the air entering through the cone around it.

The oil burns without noise and produces a beautiful lance-head flame nearly white in color. The combustion, by giving a slight excess of air, can be made absolutely smokeless, but the vessels usually operate with a slight haze issuing from the smoke pipes, as by so doing they can regulate much closer to maximum efficiency conditions than if no smoke is showing.

OIL FUEL FOR BATTLESHIPS.

The first installations fitted to battleships and, in fact, the only installations until the plans of the *Nevada* and *Oklahoma* were developed, were for boilers primarily fitted for burning coal as the regular fuel, the oil fuel being only used as an emergency aid in maintaining the required steam when it became necessary to bring coal from remote bunkers to the firerooms in use.

The same system of atomization as already described is fitted, the burners being located between the furnace doors of the boilers.

On account of the difficulty of maintaining the necessary triple balance between oil, coal and air supply, the results obtained are not as satisfactory as when burning either oil or coal alone. The first installations were rendered still more unsatisfactory by the reluctance of the boiler manufacturers to modify the designs of the furnaces and furnace fronts in order to meet the demands of the new conditions.

The necessary changes have, however, gradually been realized, and in the *Florida, Wyoming* and *Arkansas* we have practically identical conditions of air admission for the oil as exists in boilers for oil burning only, while in the *Texas* and *New York*, in addition to having this same system of air admission, the furnace volumes have been very considerably increased.

In the case of the latest design, the battleships *Nevada* and *Oklahoma*, the Department has made a radical departure, and so far as fuel is concerned, boilers and method of burning the oil, the vessels have become gigantic destroyers.

By adopting oil as the only fuel for these vessels the fireroom weights have been decreased about 300 tons, the necessary fuel weight for the designed cruising radius decreased in about the proportion of 9 to 7, the fireroom force decreased fully 50 per cent., while the total length of the ship required for boilers and firerooms has decreased from 128 feet to 66 feet.

EVAPORATION WITH OIL FUEL.

The evaporative results obtained by the use of oil fuel with mechanical atomization are quite good, as can be seen from the following table:

Name of vessel.	Minimum power.		Maximum power.		Type of cone and burner.
	Lbs. oil per sq. ft. H. S.	Water per lb. oil.	Lbs. oil per sq. ft. H. S.	Water per lb. oil.	
Trippe	.0754	12.787	.895	11.905	Normand.
Paulding	.147	10.85	.951	11.90	Normand.
Drayton	.132	12.88	.955	10.77	Normand.
Perkins	.153	12.03	.911	10.34	Fore River.
Sterett	.157	11.81	1.012	10.427	Fore River.
Walke	.0827	13.20	.79	12.94	Fore River.
McCall	.136	12.579	.785	12.31	Schütte-Koert.
Burrows	.134	12.865	.755	12.937	Schütte-Koert.
Ammen	.096	10.541	.848	11.989	Schütte-Koert.
Warrington	.196	10.046	.993	10.446	S. K.-Peabody.
Mayrant	.12	11.584	1.02	11.126	S. K.-Peabody.
Patterson	.088	12.582	.707	14.009	S. K.-Peabody.
Terry	.147	11.95	.984	11.678	Thornycroft.
Monaghan	.108	11.349	.917	11.654	Thornycroft.

From this table we see that the average evaporation from actual boiler conditions for low rates of combustion was 11.91 pounds at an average feed temperature of 155.6 degrees F. and an average boiler pressure of 238 pounds per gage, no correction for the quality of the steam being made.

For high rates of combustion these figures become 11.745 pounds, 163.5 degrees and 257.16 pounds pressure, respectively.

Estimating that the steam will be dry at the low rates of combustion and that at the high rates there will be about 3 per cent. of moisture, the evaporations under the two conditions reduced to "from and at 212 degrees" become,

For low rates of combustion, 13.23 pounds.
For high rates of combustion, 12.58 pounds.

The above results are the average under trial-trip conditions and were obtained with "Express" type boilers, which are unquestionably inferior to some other types. It is further worthy of notice that as the trial-trip crews became more accustomed to the management of oil fuel the results became better.

The results obtained with the mechanical system of atomization, burning oil under a Babcock & Wilcox boiler, with the Peabody burner and air register, as reported in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, were as follows:

Pounds oil per square foot H.S.	Pounds water evaporated per pound oil from and at 212 degrees F.	Equivalent pounds coal per square foot G. S. per hour.
.259	15.86	16.13
1.56	13.70	75.34

The boiler efficiency at the low rate of combustion figured out as 80.21 per cent., and at the maximum rate as 69.29 per cent.

CONCLUSION.

The advance of the service has not been confined solely to improvements in the main propelling engines and in the adoption of oil as a fuel, but there has been a general advance all along the line, such as in the adoption of forced lubrication to all reciprocating engines, both large and small; in the adoption of electric-driven blowers for large vessels and of turbine-driven blowers for destroyers; in improvements in condensing apparatus, feed heaters, pumps, evaporating and distilling apparatus; in fact, a close watch has been kept on every item of machinery both in its design and its operation in order that the vessels of the Navy should be maintained at the highest point of efficiency and that the dollars of the public should be spent in such a manner as would return to the public the greatest value for the money expended.

SUPERHEATERS IN MARINE BOILERS.

RESULTS OF EXPERIMENTS WITH A WATER-TUBE BOILER WITH SPECIAL REFERENCE TO SUPERHEATING.*

By HAROLD E. YARROW, Associate Member.

The author's firm, having recently tried some important experiments with a water-tube boiler, it was thought that the results would be of interest to the institution, and he therefore ventures to bring them before this meeting. Being convinced that superheating is one of the directions in which advances in marine engineering will be made, a water-tube boiler was constructed which was fitted with a special form of superheater. The objections which have hitherto been raised to superheating for marine work are:

1. Owing to the dryness of the steam, oil for the internal lubrication in reciprocating engines becomes a necessity, and the oil, finding its way into the boiler, leads to trouble.

2. The probability of burning the superheater when the passage of the steam through it is suddenly reduced or stopped.

By the introduction of turbines the difficulty of lubrication does not occur, and with regard to burning the superheater tubes, the arrangement adopted by the author's firm avoids this risk.

The boiler with which these tests were made was of the Yarrow type, and was fitted up in an experimental shop, which is equipped with the necessary plant for making very complete tests. Throughout the experiments oil fuel only was used; and as it is possible with oil to maintain steady and uniform working conditions, very accurate results were obtained, which would not have been possible with coal, in which case irregularity of stoking and other sources of discrepancy occur. During the experiments careful records were taken of the oil consumed, the water evaporated, steam pressure, temperature of the superheated steam, and the temperature of the gases at various points during their passage past the boiler tubes.

As is well known, superheating is very largely adopted in land installations, and in locomotives it is being rapidly introduced. From information kindly given by the locomotive superintendents of the main railway lines in this country, it appears that the economy realized in locomotives due to superheating averages fully 20 per cent. in fuel consumption, and rather more in water consumption. It may perhaps surprise many marine engineers to know that on the Great Western Railway alone no fewer than 500

* Paper read before the Institution of Naval Architects, March 28, 1912.

locomotives are now running fitted with superheaters. Taking the average of several land turbine installations, there is found by superheating to 100 degrees F. to be a saving in consumption of fuel of from 8 to 10 per cent., and in steam consumption from 10 to 12 per cent. The reduction in steam consumption is specially important for marine work, as it enables a reduction to be made in the size and weight of the condensers, air pumps, circulating pumps and feed pumps, and probably of the distilling plant. Independent of the gain directly due to superheating, the risk of water passing into the turbine from any cause whatever is reduced, and the fear of damage in consequence of water causing the stripping or cutting of the blades is diminished, and any additional cost of upkeep of the superheater will doubtless be fully balanced by the diminished risk of injury to the turbine blades by the action of water when using saturated steam.

Turning now to the design of the boiler and superheater with which the various experiments were carried out, and referring to Fig. 1, which shows

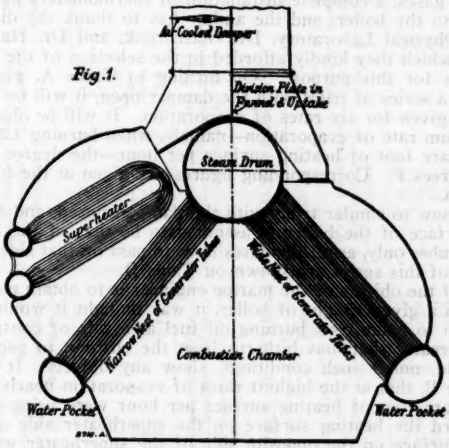


Fig. 1.

a cross section of the boiler, it will be seen that it consists of a top steam collector, as usual, and two lower water pockets. On the left-hand side the superheater is shown, and it will be observed that on this side of the boiler there are fewer rows of generator tubes than on the other side, where there is no superheater, it being thought desirable that the total heating surface and the resistance to the gases on both sides of the boiler should be approximately the same. The total heating surface of the boiler was 6,700 square feet, of which 1,265 square feet consisted of superheating surface; the total heating surface on the superheater side of the boiler was 3,453 square feet, and on the other side 3,247 square feet.

The superheater consisted of a number of U tubes expanded into two longitudinal collectors, small doors being fitted so that access to the tubes could be obtained when required. The leading feature of the arrangement is that the superheater is placed on one side of the boiler only, and a damper is fitted in the up-take on the same side, as shown on the diagram. If this damper is closed, the whole of the gases are deflected towards the opposite side of the boiler, and no heated gases pass the superheater, the object being that if the main engine should be suddenly eased or stopped, or when raising steam, the superheater may be shut off, so as to prevent the tubes being damaged, or the steam being superheated to an excessive

extent, owing to there not being sufficient circulation of steam. In this way one objection to the introduction of superheating for marine installations is overcome.

A further advantage of this arrangement is that when the consumption of steam is suddenly reduced or stopped, not only does the damper prevent the superheater tubes from being burnt, but it also greatly diminishes the output of the boiler at the time when a reduced supply of steam is wanted, because only about one-half of the heating surface comes into contact with the hot gases. To avoid the possibility of the damper getting distorted through overheating, it is provided with a hollow spindle, to which air is admitted, and which passes from thence between the two plates of the damper, escaping at the edges, and thus keeping the damper cool. This arrangement of damper has proved thoroughly successful under the most trying conditions.

In order to carefully measure the temperatures of the superheated steam and of the gases, a complete installation of thermometers and pyrometers was fitted to the boiler, and the author has to thank the director of the National Physical Laboratory, Dr. Glazebrook, and Dr. Harker, for the assistance which they kindly afforded in the selection of the most reliable instruments for this purpose. On turning to Table A, giving the particulars of a series of trials with the damper open, it will be seen that the results are given for six rates of evaporation. It will be observed that at the maximum rate of evaporation—namely, when burning 1.237 pounds of oil per square foot of heating surface per hour—the degree of superheat was 93 degrees F. Corresponding figures are given at the lower rates of evaporation.

Passing now to similar trials with the damper closed, and for which the heating surface of the boiler is assumed to be that of the large nest of generator tubes only, as all the gases have to pass on that side of the boiler the results of this series are shown on Table B.

As one of the objects of the marine engineer is to obtain more and more steam from a given weight of boiler, it was thought it would give useful information to make tests burning oil fuel at a rate of consumption considerably greater than has hitherto been the custom, to ascertain if the boiler would, under such conditions, show any defects. It will be seen from Table B, that at the highest rates of evaporation nearly 2 pounds of oil per square foot of heating surface per hour were being consumed, if we disregard the heating surface on the superheater side of the boiler. Thus the surface on the opposite side of the superheater was subject to the heating effect of all the gases plus half the radiation. Every part of the boiler withstood the severe test, and trials burning this quantity of oil were made on several occasions.

The results of other experiments indicate that in a properly-designed boiler of the type under discussion it is possible to burn, without injury to the boiler, 2 pounds of oil per square foot of heating surface per hour.

Since these experiments were carried out the official trials have taken place with one of the destroyers built by Messrs. Yarrow & Co., Limited, H.M.S. *Archer*, in which boilers fitted with superheaters were provided. The result of these trials showed that the gain expected was fully realized, and on the full-speed trial the degree of superheat at the turbines was 94 degrees F.; the shaft horsepower developed was slightly over 18,500, which compares with about 17,000, which is the shaft horsepower the builders expected had the boilers been of the usual type. The mean speed obtained on the six runs on the measured mile at Skelmorlie was 30.9 knots, and the mean speed for eight hours 30.3 knots, the contract speed being 28 knots.

One point to which Mr. Charles Merz (to whom the author is greatly indebted for much valuable information) has drawn attention is the necessity, with superheated steam, of efficiently covering the high-pressure portions of the turbine cylinder with non-conducting material, because the

TABLE A.—TRIALS WITH DAMPER OPEN.

Steam pressure, lbs. per sq. in.	Superheat, in degrees F.	Air pressure, in inches of water.	Pounds of water evaporated per hour.	Pounds of oil fuel burnt per hour.	From and at 212 degrees F.		Pounds of oil fuel burnt per square foot of heating surface per hour.	Temperature of feed water, in degrees F.	Temperature between small nest of generator tubes and superheater, in degs. F.	Temperature of uptake, in degrees F.	
					Water per pound of oil.	Water per square foot of heating surface.				Above superheater.	Above large nest of generator tubes.
242.0	93.5	5.0	94,659	8,286	14.6	18.0	1.237	58.0	1,121	828	887
243.0	93.0	3.16	76,021	6,454	15.0	14.4	0.9635	63.5	926	698	727
243 7	82.5	2.44	68,387	5,695	15.2	12.9	0.850	63.5	903	685	688
242 8	61.1	1.7	46,041	3,630	15.9	8.6	0.542	64.0	647	536	551
241.8	31.0	0.998	20,059	1,540	16.1	3.7	0.230	62.2	481	432	448
242.2	20.75	0.625	8,478	649	16.1	1.55	0.096	63.5	465	409	416

Heating surface { Large nest of generator tubes.....3,247 square feet; Small nest of generator tubes.....2,188 square feet; Superheater.....1,265 square feet } 6,700 square feet, total.

On these trials the heating surface is taken as the total heating surface of 6,700 square feet.

TABLE B.—TRIALS WITH DAMPER SHUT.

Steam pressure, pounds per square inch.	Air pressure, inches of water.	Pounds of water evaporated per hour.	Pounds of oil fuel burnt per hour.	From and at 212 degrees F.		Pounds of oil fuel burnt per square foot of heating surface per hour.	Temperature of feed water, in degrees F.	Temperature of uptakes, in degrees F., above large nest of generator tubes.
				Water per pound of oil.	Water per square foot of heating surface.			
242 0	4.85	68,648	6,287	13.25	25.66	1.936	61.0	913
242.25	3 97	57,693	5,065	13.64	21.6	1.56	60.0	843
242.4	2 491	44,950	3,504	15.3	16.5	1.09	60.3	673
242.5	1.46	31,481	2,473	15.4	11.75	0.76	63.5	603

Heating surface { Large nest of generator tubes.....3,247 square feet; Small nest of generator tubes.....2,188 square feet; Superheater.....1,265 square feet } 6,700 square feet, total.

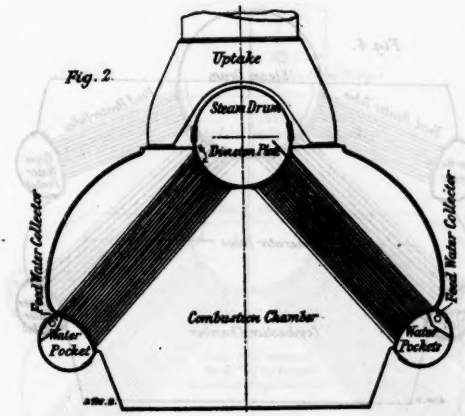
On thess trials the heating surface of boiler is taken as heating surface of large nest of generator tubes = 3,247 square feet.

metal on the inside of the cylinder in contact with the steam becomes hotter than that on the outside, especially at the edges of flanges and ribs. The inside tends to expand, and this expansion is resisted by the colder metal on the outside; consequently, if the temperature difference is great enough, the metal will be distorted and perhaps strained beyond its elastic limit. For this reason the design of the ribs should be carefully considered, the thickness of metal throughout the structure being kept as uniform as possible.

These experiments have also been the means of pointing out another important improvement in the Yarrow boiler, and Fig. 2 has been prepared, which illustrates the ordinary type of boiler without superheater, from which it will be seen that the last two rows of tubes farthest from the fire are partitioned off for the feed water to ascend. When working the boiler *at high rates of evaporation*, it was found that notwithstanding all possible precautions, even with turned rivets and carefully reamered holes, it was impossible to prevent the riveted seam of the water pockets from leaking. It was found the pocket was sometimes hot and sometimes cool; indeed, in places sufficiently cool to be able to bear one's hand on it. These trials were frequently repeated with the same result, and ultimately the cause was found out. The fact was that the suction down the tubes which were in close proximity to the feed-heating tubes was so great that the cool feed water which had passed up the feed-heating tubes was instantly drawn down into the water pocket without having had time to mix with the hotter water in the upper chamber, as indicated by the arrow in Fig. 2. This action took place intermittently, and the water pockets locally changed their temperature, one portion of the water pocket being one minute hot, and another minute comparatively cool, dependent upon the working of the feed pump. The strain thus thrown on the metal of the water pockets by this short-circuiting of the feed was evidently severe, and resulted in the leakage of the seams. Having discovered the cause of the trouble, it was not difficult to find a remedy. It was found that by simply placing a longitudinal partition plate in the upper chamber, so as to avoid the short-circuiting of the feed, all difficulties disappeared. This plate is shown on the right-hand side of the diagram only, but in practice would, of course, be fitted on both sides. The same trouble has before been met with, especially abroad, but the true cause was never suspected; it was generally put down to inferior workmanship. Even when no trouble is experienced it is evident that serious and undesirable strains must at times be taking place, which may in the end lead to the failure of the metal, due to constant fatigue. By the fitting of this partition plate, however, all such strains are eliminated.

Fig. 3 has been prepared to show the positions of the pyrometers. Table C, Fig. 8, shows the temperature of the gases at various points in the boiler. The vertical lines correspond to the position of the pyrometers as shown. The upper curve indicates the gas temperatures at a rate of evaporation of 16 pounds of water per square foot of heating surface, and the lower curve represents the gas temperatures at a rate of evaporation of slightly over 3 pounds per square foot of heating surface. The horizontal lines represent temperatures, and the line B C represents the temperature of the steam at 200 pounds pressure, and the line E F the temperature of the air-pump discharge taken at 78 degrees F.

It will be observed that a very great drop in temperature takes place during the passage of the gases through the first rows of tubes, showing the large proportion of heat that is taken out of the gases by these tubes. Also, it will be observed that there is a sudden drop in the temperature at A and A′—that is, where the gases pass through the last rows of tubes. This is due to the fact that the cold feed water (which enters a portion of the water pocket) abstracts a greater amount of heat from the gases in ascending the two outside rows of tubes than would be the case if these tubes were full of water at the temperature of the steam.



Referring to Table C, given in diagram form, it will be seen that the temperature of the gases at the point A′—*i. e.*, just prior to the gases passing the feed-heating tubes—is about 550 degrees F., and the temperature of the steam at 200 pounds pressure is 388 degrees F., a difference of only 162 degrees, whereas the temperature of the air-pump discharge of 78 degrees F. gives a difference of 472 degrees. This clearly shows the gain due to this system of feed heating and the desirability of extending it, which can be effected by having separate water collectors and feed-heating tubes apart from the main water collectors and main generator tubes—*i. e.*, there would be two water collectors on each side of the boiler, the tubes connected to the top one acting as a feed heater, and such a design of boiler is shown in Fig. 4.

It should also be pointed out that there is supplementary and an important advantage in this feed heating—namely, that any grease or sediment that comes over with the feed is deposited in these tubes, which are not subject to fierce heat, rather than in those near the fire, which are exposed to the intense radiation of the furnace, and thus the life of the boiler is prolonged. With the introduction of oil fuel some such arrangement is the more necessary, because it has been found that the oil heaters

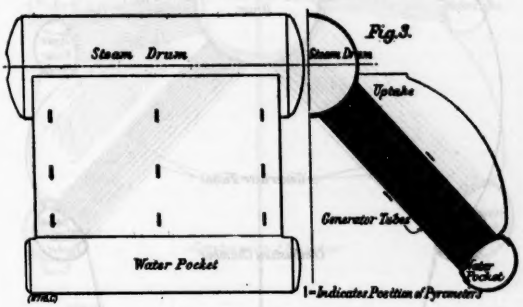
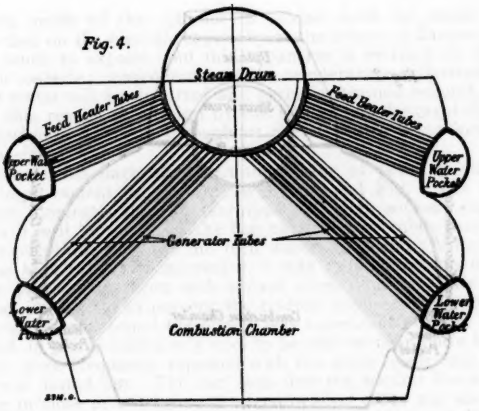


Fig. 4.



leak, with the result that oil mixes with the steam and passes ultimately into the boiler.

As the author thought it would interest the meeting to indicate some of the arrangements for superheating and feed heating which may be adopted with a view to still further improving the results in connection with such a boiler as the one being dealt with, he begs your reference to three illustrations.

Fig. 5, it will be seen, shows the superheating tubes united to the steam drum, and to a steam receiver sufficiently large for a man to enter, the tubes being expanded at both ends. In this arrangement all the tubes are

Fig. 5.

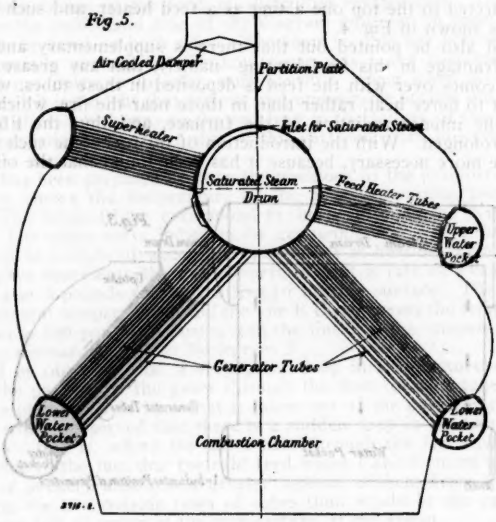


Fig.6.

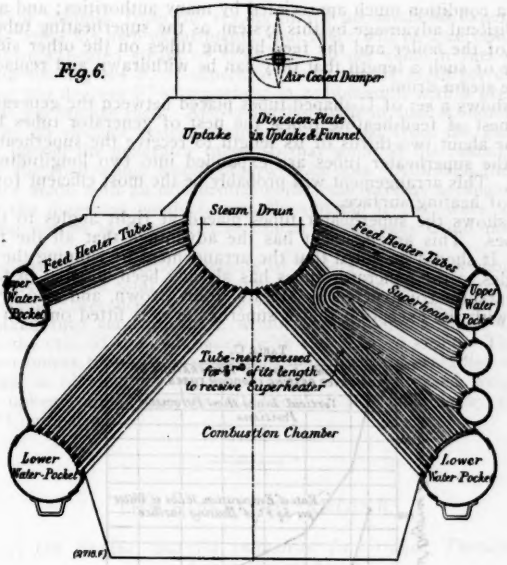
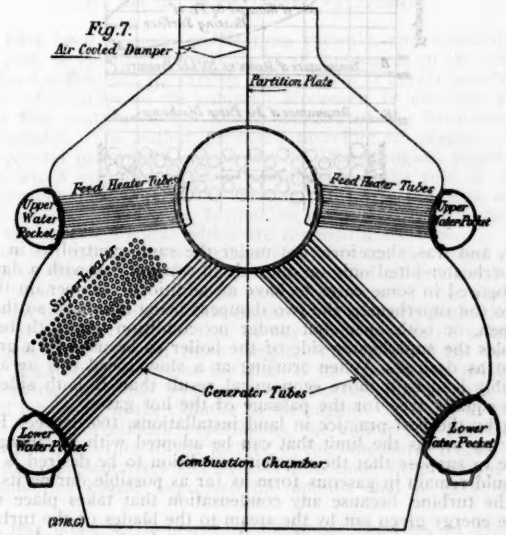

Air Cooled Damper
Uptake
Division Plate in Uptake & Funnel
Steam Drum
Feed Heater Tubes
Feed Heater Tubes
Upper Water Pocket
Upper Water Pocket
Superheater
Tube next recessed for ⅛th of its length to receive Superheater
Combustion Chamber
Lower Water Pocket
Lower Water Pocket
(2718.F)

Fig.7.

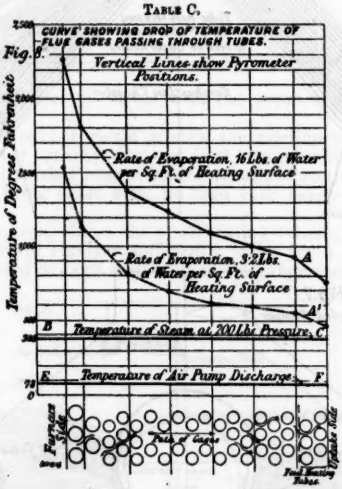

Air Cooled Damper
Partition Plate
Feed Heater Tubes
Feed Heater Tubes
Upper Water Pocket
Upper Water Pocket
Superheater
Generator Tubes
Lower Water Pocket
Lower Water Pocket
Combustion Chamber
(2718.G)

straight, a condition much appreciated by many authorities; and also there is an additional advantage by this system, as the superheating tubes on the one side of the boiler and the feed-heating tubes on the other side of the boiler are of such a length that they can be withdrawn and replaced from inside the steam drum.

Fig. 6 shows a set of U-shaped tubes placed between the generator tubes and the nest of feed-heating tubes, the nest of generator tubes being recessed for about two-thirds of its length to receive the superheater. The ends of the superheater tubes are expanded into two longitudinal steam receivers. This arrangement will probably be the most efficient for a given quantity of heating surface.

Fig. 7 shows the superheater tubes placed at right angles to the generator tubes. This arrangement has the advantage that all the tubes are straight. It should be noted that the arrangement of running the tubes at right angles to the generator tubes has already been adopted in boilers of certain warships constructed by Messrs. John Brown, and also by Messrs. Yarrow, with the exception that a superheater was fitted on both sides of

TABLE C.


CURVE SHOWING DROP OF TEMPERATURE OF FLUE GASES PASSING THROUGH TUBES.
Fig. 8.
Vertical Lines show Pyrometer Positions.
Temperature of Degrees Fahrenheit
Rate of Evaporation, 16 Lbs. of Water per Sq. Ft. of Heating Surface
Rate of Evaporation, 5·2 Lbs. of Water per Sq. Ft. of Heating Surface
A
A'
B
Temperature of Steam, at 200 Lbs. Pressure, C
E
Temperature of Air Pump Discharge
F
Furnace Side
Path of Gases
Uptake Side
Feed Heating Tubes

the boiler, and was, therefore, not under the same control as in the case of the superheater fitted only on one side in conjunction with a damper.

It is proposed in some cases to have an additional damper on the opposite side to the superheater, the two dampers being arranged so that either can be open, or both open, but under no condition can both be closed. This enables the superheater side of the boiler to be used to a greater or less extent as desired. When cruising at a slow speed this arrangement may possibly lead to a more economical result than if both sides of the boiler are equally free for the passage of the hot gases.

Judging by the best practice in land installations, 100 degrees F. superheat is by no means the limit that can be adopted with advantage. It is reasonable to suppose that the requisite condition to be desired is that the steam should remain in gaseous form as far as possible during its passage through the turbine, because any condensation that takes place must diminish the energy given out by the steam to the blades of the turbine, also if the steam remains in gaseous form the steam friction is reduced.

The author submits, from the results of the accumulated experience of others, and from his own experiments, that there will be a certain gain by the use of superheated steam of from 8 to 10 per cent. in fuel economy when using 100 degrees F. of superheat, and from 11 to 13 per cent. gain when using 150 degrees F. of superheat in combination with a pressure of 200 pounds per square inch. Also a further gain in fuel economy can be obtained by an efficient system of heating the feed from the gases after they have passed the generator tubes, so that some of the remaining heat should be absorbed which would otherwise be lost. Although the experiments were made with an oil-burning boiler, the various arrangements as shown in the diagrams would be equally suitable if coal were used, and there is no reason to suppose that similar advantages could not be obtained.

With regard to the relative weights of boilers with and without superheaters, provided the total heating surface were the same, there would be no appreciable difference. If, however, the complete machinery installation is taken into account, there would probably be a small saving in weight in the case of the installation with superheated steam. The author therefore submits that in the propelling machinery of warships improved results can be obtained by superheating, without increased weight, cost, space or upkeep, and a further extension of feed heating by the waste gases.—" Engineering."

THE DIESEL OIL ENGINE.

*The Diesel Oil Engine, and Its Industrial Importance, Particularly for Great Britain.**

By Dr. Rudolph Diesel, of Munich.

There have been so many publications recently, and especially during the past year, in technical periodicals of all languages, on the construction of the Diesel engine and its various types, that it is hardly possible to give any fresh information on the subject. Moreover, an excellent paper was read only last summer at the Zürich meeting of the Institution by Mr. J. F. Schubeler. The author proposes, therefore, to discuss only questions of general importance concerning the Diesel engine, especially those questions which are brought into prominence in the title of the paper, and to consider them only as the starting point of a thorough and stimulating discussion. He desires, further, to be exonerated if statements are made in the paper on points which are common knowledge.

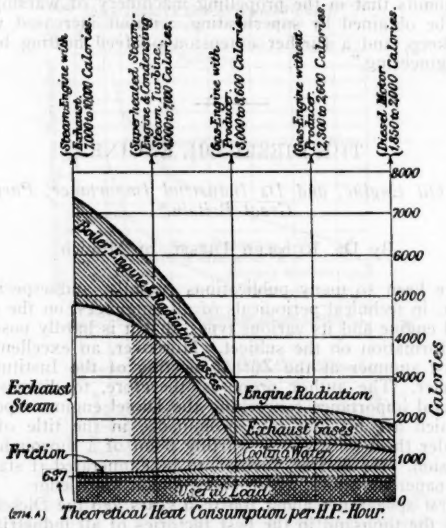
Since its first appearance about fourteen years ago the Diesel engine has been built by the thousand in the best factories of all industrial countries, and has been set up in the most remote corners of the world. It has been proved to be a most reliable engine when properly built, the working of which is quite as safe as that of any other system of prime mover; and in general it is even more simple, since it does not require any auxiliary apparatus, and since the fuel in its natural and original form, without having previously undergone any transforming process, is directly converted into work in the cylinder of the engine. As early as 1897, when, after four years of difficult experiments, the author had put the first engine into working order in the factory of the Augsburg Works, numerous engineering representatives and experts who came from various countries to examine this engine expressed the opinion that it gave better heat

* Paper read before the Institution of Mechanical Engineers on Friday, March 15.

utilization than any known kind of heat engine. From experience gained subsequently by working many engines, by gradual improvements in the construction and manufacture, and by increasing the sizes, the results have been still further improved, and today the thermal or indicated efficiency reaches 48 per cent. in this engine, and the effective or brake efficiency reaches, in some cases, 35 per cent. of the heat value of the fuel.

Fig. 1 shows the heat utilization for 1 brake horsepower hour in the different kinds of prime movers known today. Science and technical knowledge are making continuous progress, and the time will come when even these figures will be exceeded; but, with our present scientific knowledge, any considerably higher efficiency in the process of transforming heat into mechanical work is not obtainable. Further progress seems only possible with some other method for transforming heat into work; this would mean an entirely new principle, which, however, one cannot speculate upon in the present state of science.

Fig. 1. HEAT CONSUMPTION OF DIFFERENT HEAT ENGINES PER B.H.P.-HOUR.



The Diesel engine is therefore the engine which converts the heat of the natural fuel into work in the cylinder itself, without any previous transforming process, and which utilizes it as far as the present standard of science permits; it is therefore the simplest and, at the same time, the most economical prime mover.

These two facts explain its success; it lies in the new principle of the internal-working process, and not in constructional improvements or alterations of older types of engines. There is no doubt that the careful working out of all the constructional details also plays a great part in the practical success of the Diesel engine, as in any other; but they are not the essential points, and, above all, they do not constitute the great importance of this engine to the world's industry.

A further reason for this importance is that the Diesel engine has

broken the monopoly of coal, and has solved the problem of using liquid fuel for power production in its simplest and most general form. It has become for all liquid fuels what the steam engine and gas engine are for coal, but in a much simpler and more economical way. The truth of this statement was strikingly proved at the Turin Exhibition of last year. At this exhibition, in the large Machinery Hall, a steam turbine and a large Diesel engine, both made by Franco Tosi, of Milan, and set up on the same stand, were worked together with the same liquid fuel. The boilers belonging to the plant were fitted with Körting nozzles for burning crude oil. The difference between the two plants was therefore this: for the working of the steam engine the whole boiler plant, with its chimney, fuel-supply apparatus, purification plant for feed water with feed pumps, extensive steam pipes, condensation plant with water pumps, and an enormous water consumption, had to be provided, with the final result of consuming 2½ or more times the fuel per horsepower required by the Diesel engine standing beside it. The latter, being an entirely independent engine, without any auxiliary plant, took up its crude fuel automatically, and consumed it direct in its cylinders without any residue or smoke. A better proof can hardly be imagined, even for the non-technical man, that, except in special cases, the steam engine cannot compete economically with the oil engine, and from this point of view the power plant in the Machinery Hall at Turin must be looked upon as marking a historical event. It is hardly possible for a country which produces no coal, like Italy, to develop a great industry based on the steam engine, and this is one reason for the exhibition at Turin of about thirty Diesel engines of various types and sizes, and made in different countries.

Thus the Diesel engine has doubled the resources of mankind as regards power production, and has made new and hitherto unutilized products of Nature available for motor power. The Diesel engine has thereby exercised a far-reaching influence on the liquid-fuel industry, which is at the present time improving more rapidly than was previously conceivable. This is not the place to discuss the matter in detail, but the author wishes to mention that, owing to the interest which petroleum producers have taken in this important question, new petroleum sources are continually being developed and new oil districts discovered. Moreover, it has been proved by recent geological researches not only that there is probably on the globe as much, or perhaps even more, liquid fuel than coal, but also that it is more conveniently distributed as regards its geographical position. Fig. 2. These facts, which are indisputable nowadays, have gradually silenced those who objected to too great a development of the Diesel engine for fear of insufficient stores of liquid fuel. Any such anxiety may be relieved by the fact that the world's production of crude oil increases at present 3½ times more quickly than the production of coal, and that the ratio of increase itself is steadily getting higher. Further, that 40 per cent. of the present production of mineral oil is already sufficient to supply the whole of the naval and mercantile fleets of the world with power, if they were worked by Diesel engines: also that with the world's present production the number of Diesel engines now working could be increased about a hundredfold.

It may thus safely be asserted that, with the continual development of new oil districts, the production of mineral oil will increase much more quickly than the demand for newly-built engines. It is therefore not surprising that, in the last yearly report of the Shell Transport Company, attention was called to the fact that the oil consumption in these engines did not nearly equal the production, and that the company has to look out for other markets to dispose of their superfluous stock. That the auxiliary industries of petroleum production are also considerably influenced is shown by the great increase which the transport industry for liquid fuel has experienced in recent times, especially the great develop-

ment of tank vessels which are, or will be, mostly driven by Diesel engines.

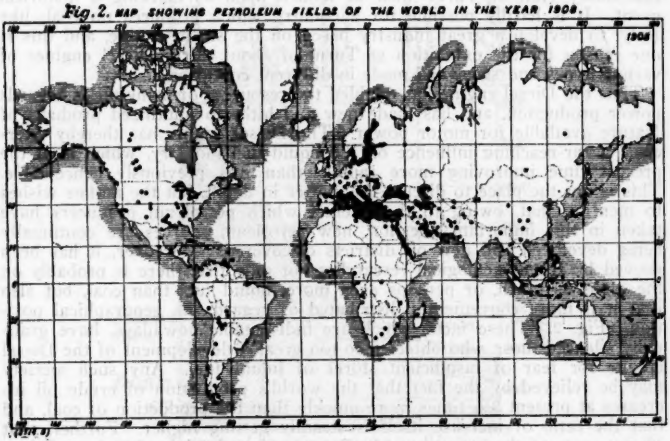
But with all this the influence of the Diesel engine in the world's industry is not exhausted. As early as the year 1899 the author utilized in his engine the by-products of coal distillation and coke plants, such as tar and creosote oils, with the same satisfactory results as with natural liquid fuels, but at that time the quality of these oils was generally too inferior for their use in the Diesel engine, and it was, moreover, subject to continual variations.

The difficulties were then chiefly the following:

1. Muddy deposits of solid hydrocarbon, especially naphthalenes, which made the working of the fuel pumps difficult, filled up the pipes and nozzles, and formed a hard crust at the nozzle mouths. These solid hydrocarbons also made higher ignition temperatures necessary.

2. Continuous change in quality and composition of the crude, uncleaned tar oils; with each cask fresh variations appeared, even in cases where the works guaranteed the use of the same coal and the carrying out of the same distilling process, so that it was impossible to make scientific observations for drawing any definite conclusions or making logical experimental arrangements. The characteristics of crude tar oils were not then exactly known, even to the producers; for instance, nobody imagined that differences in the distilling temperature and variations in the nature

Fig. 2. MAP SHOWING PETROLEUM FIELDS OF THE WORLD IN THE YEAR 1908.



and position of the retorts gave entirely different tar products even when the same coal was used. It is only in recent years that the chemical industries interested in the matter have, by improved methods of fractioning and refining, combined with more careful selection of the material, succeeded in supplying fuel of a constant and regular quality, without the drawbacks of the crude tar oils used previously. These products—the tar and tar oils—are thus today definitely brought into the sphere of activity of the Diesel engine.

In Appendix I is given a table showing the application of various fuels for Diesel engines, while Appendix II contains specifications for tar oils for Diesel engines, and Appendix III gives the properties of tar.

From what has been just stated it will be seen that the Diesel engine is having an increasing influence on two other industries—the manufacture of gas and coke—the by-products of which have become so impor-

tant for power production that an enormous business is at present connected with them.

It is especially noteworthy that every town gas works with a modern installation, and every coke works, can be completed with an electric-power generating plant by using its tars. This will have an excellent effect on many municipal and national works. It would take too much time to enter into the details of this question, but one fact stands out clearly in this connection—namely, that coal, which seemed to be most threatened by the liquid fuels, will, on the contrary, gain a new and wider ground of application through the Diesel engine. As tar and tar oils are from three to five times better utilized in the Diesel engine than coal in the steam engine, a much better and more economical utilization of coal is obtained if, instead of being burned under boilers on grates in a wasteful way, it is first transformed into coke and tar by distillation. Coke is used in metallurgical and other general heating purposes; from a part of the tar the valuable by-products are first extracted, and undergo further processes in the chemical industry, whilst the tar oils and combustible by-products, and a great part of the tar itself, are burned in the Diesel engine under extraordinarily favorable conditions.

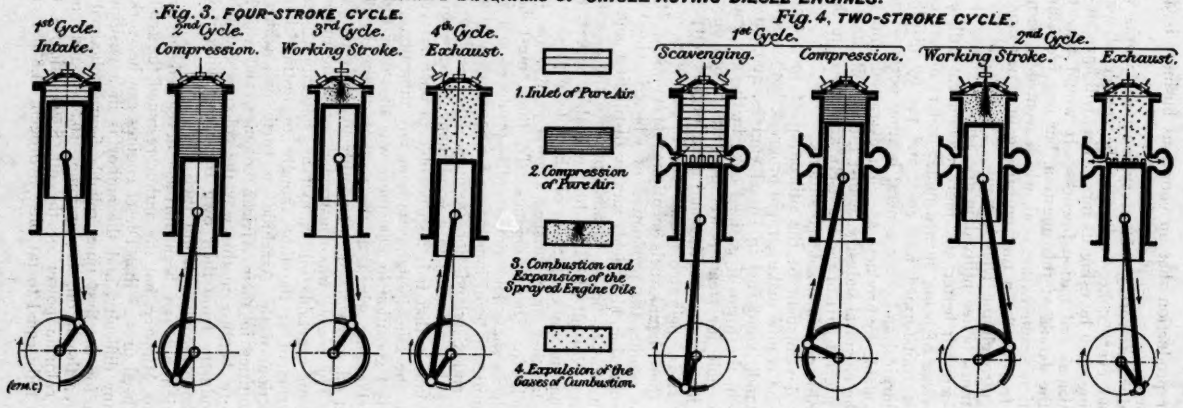
The proper development of the utilization of fuel, which has already been started, and is now making rapid progress, is therefore the following: On the one hand, liquid fuel in Diesel engines, and, on the other hand, gas fuel, also in the form of gasified coke in the gas engines; solid fuel as little as possible for steam-power generation, but only in the refined form of coke for all other heating and metallurgical purposes.

The list of fuels applicable to the Diesel engine is not, however, exhausted with these liquid fuels mentioned previously. It is known that brown coal or lignite, the production of which is about 10 per cent. of the pit-coal production, also yields tars by distillation, and these tars when distilled, when worked up on paraffin, produce the so-called paraffin oils as by-products. Not all kinds of brown coal, however, are suitable for this process; but, in any case, these oils have been produced in such quantities that so far they have satisfied a good part of the German demand for liquid fuels for Diesel engines. To these must be added other products, such as shale and similar oils—produced not in great, but in sufficient quantities to be of importance for power generation. Some countries—France and Scotland, for instance—possess them in considerable quantities, and they are used in a good many Diesel-engine plants. It is not generally known that it is also possible to burn fat vegetable oils and animal oils in the Diesel engine without any difficulty.

At the Paris Exposition in 1900 there was shown by the Otto Company a small Diesel engine which, at the request of the French Government, ran on peanut oil, and worked so smoothly that only very few people were aware of it. The engine was constructed for using mineral oil, and was then worked on vegetable oil without any alterations being made. The French Government at the time thought of testing the applicability to power production of the Arachide, or earth nut, which grows in considerable quantities in their African colonies, and which can be easily cultivated there, because in this way the colonies could be supplied with power and industry from their own resources, without being compelled to buy and import coal or liquid fuels.

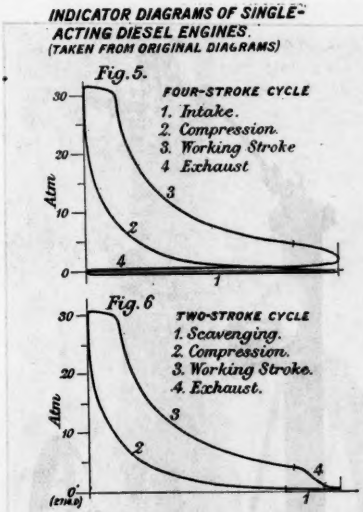
This question has not been further developed in France owing to changes in the Ministry, but the author resumed the trials a few months ago. It has been proved that Diesel engines can be worked on earth-nut oil without any difficulty, and the author is in a position to publish, on this occasion for the first time, reliable figures obtained by tests: Consumption of earth-nut oil, 240 grammes (0.53 pound) per brake horse-power-hour; calorific power of the oil, 8,600 calories (34,124 B.T.U.) per kg., thus fully equal to tar oils; hydrogen, 11.8 per cent. This oil is

WORKING DIAGRAMS OF SINGLE-ACTING DIESEL ENGINES.



almost as effective as the natural mineral oils, and as it can also be used for lubricating oil, the whole work can be carried out with a single kind of oil produced directly on the spot. Thus this engine becomes a really independent engine for the tropics.

Similar successful experiments have also been made in St. Petersburg with castor oil; and animal oils, such as train oil, have been used with excellent results. The fact that fat oils from vegetable sources can be used may seem insignficant today, but such oils may perhaps become in



course of time of the same importance as some natural mineral oils and the tar products are now. Twelve years ago the latter were not more developed than the fat oils are today, and yet how important they have since become. One cannot at present predict what part these oils will play in the Colonies in the future. In any case, they make it certain that motor power can still be produced from the heat of the sun, which is always available for agricultural purposes, even when all our natural stores of solid and liquid fuels are exhausted.

Historical Summary.

The author thinks that a summary of the whole development of the Diesel engine, and of the general points connected therewith, with illustrations of a few engines which mark stages in its evolution, may be of interest to the members. Several of these have already been published separately in the technical Press, but the series, as a whole, in its historical connection is quite new, and a certain number of the photographs have not been previously published. Figs. 3 and 4 show small illustrations of the principal movements in the four-stroke and two-stroke cycle engines, with the corresponding indicator diagrams in Figs. 5 and 6, because these will constantly be referred to in the paper.

FOUR-STROKE CYCLE ENGINE.

Vertical Stationary Engines.—The first experimental Diesel engine, Fig. 7, constructed in 1893, had the piston fitted with a piston rod and external crosshead, the cylinder having no water jacket; the cam shaft was arranged very low, and the valves were actuated by means of long rods. The starting storage chamber consisted of a wrought-iron pipe with riveted flanges, and there was no air-supply pump, the fuel being injected directly.

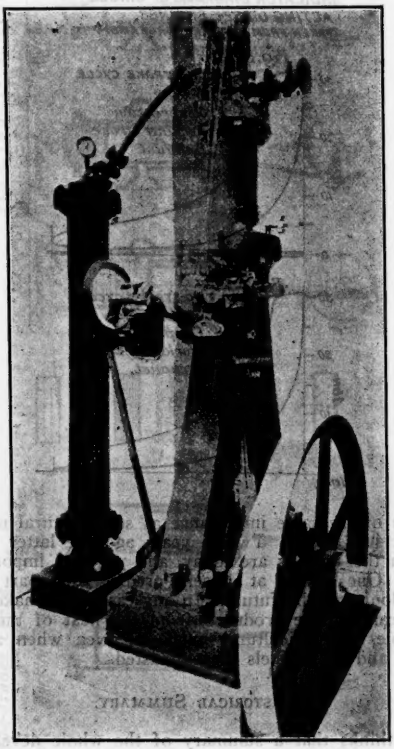


FIG. 7.—FIRST EXPERIMENTAL DIESEL ENGINE, 1893.

A later pattern (Fig. 8), built in 1895–6, had a similar base to that shown in Fig. 7, but it had a water-jacketed cylinder, and the cam shaft was placed higher up. But the most important difference from the old pattern was in the air-supply pump, the necessity for which was only recognized after several years' experimenting, as without it a smokeless combustion could not be effected. This air pump is single-acting, but the author previously used a special vertical compound air pump driven from a transmitting shaft.

The first French and Belgian engines were nearly of the same type as that shown in Fig. 8, but had no air pump; they were also of better and more compact construction.

The first reliable Diesel engine, Fig. 9, of 18 horsepower, was finished in 1897 at Augsburg, after about four years' laborious experimenting. It was a vertical engine having the piston connected to an external crosshead and worked on the four-stroke cycle. The illustration shows the engine

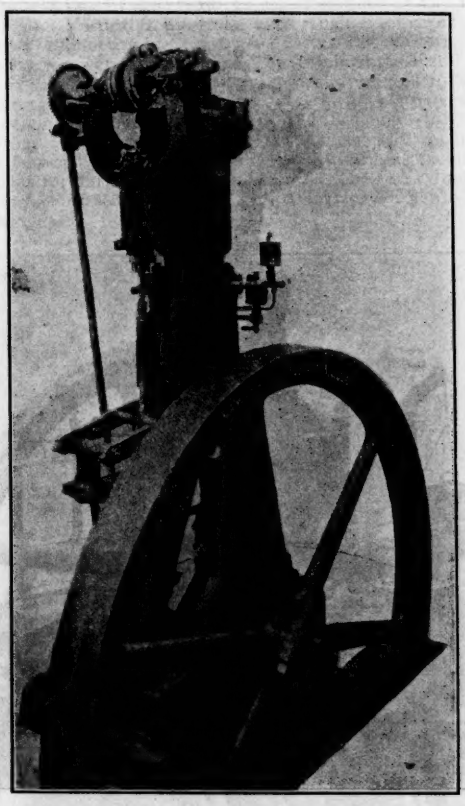


FIG. 8.—LATER EXPERIMENTAL PATTERN, 1895-6.

with the testing brake attached, and with the other testing apparatus exactly in the state in which it was used by the numerous commissions of engineers and experts who came from different countries to examine the engine, as mentioned earlier in this paper. This type was for about ten years the exclusive and almost stereotyped pattern for all Diesel engines, which were built in various countries.

In the following year, 1898, the first single-cylinder engine of 20 to 25

horsepower was built at the Augsburg Works. This engine had almost all the characteristic details of the experimental engine just mentioned, the only difference being that the experimental engine had the cylinder connected to the base by an inclined column in front, whilst the latter pattern had the well-known A-frame which the author employed on his first experimental engines. No further alterations have been made. The

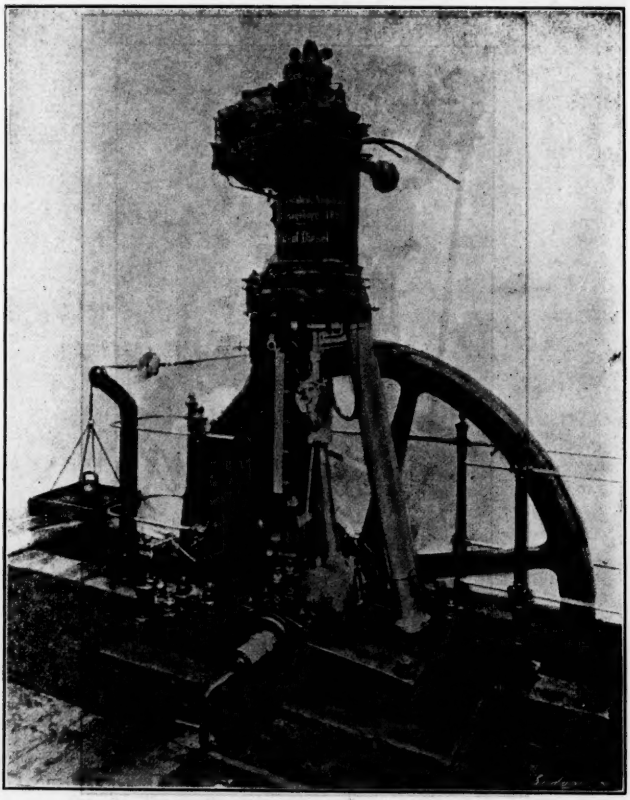


FIG. 9.—FIRST COMPLETE ENGINE, 18-HORSEPOWER, 1897; SHOWING BRAKE AND OTHER TESTING APPARATUS.

engine had still the external crosshead and guides, and the petroleum pump was actuated by the cam shaft in exactly the same way as in the experimental engine, Fig. 9. The air pump was cast on the base in exactly the same way and was driven in both cases by rocking beams from the crosshead. The lubrication of the cylinders was effected by means of Mollerup appliances, and the valve rods, the regulator, and all the details

were identical in both engines. Except for alterations in unimportant details of construction, the only changes since made were that the dimensions and the number of cylinders were enlarged.

The two-cylinder engine of 60–76 horsepower was made in 1899, in which all the details of Fig. 9 are still to be recognized. The only alteration which was made in the year 1901 was the abandonment of the external crosshead and adoption of the trunk piston shown in Fig. 10. A comparison of this engine with that illustrated in Fig. 9 shows that, with the exception of the omission of the crosshead, no alterations of any importance have been made. Vertical four-stroke cycle engines of from 10 to 250 horsepower per cylinder were constructed after this pattern, and units up to 1,000 horsepower were obtained by combining several cylinders. These engines ran at comparatively low speeds, from 160 to 200 revolutions, according to their size, and were of very heavy construction. Their weight was originally from 280 kg. to 350 kg. (617 pounds to 771 pounds) per horsepower, and later from 240 kg. to 300 kg. (529 pounds to 661 pounds) per horsepower. This type of engine was used exclusively as a stationary plant for various industrial purposes. The two-cylinder M.A.N. engine of this type of 250 horsepower, or 125 horsepower per cylinder, was built in 1902.

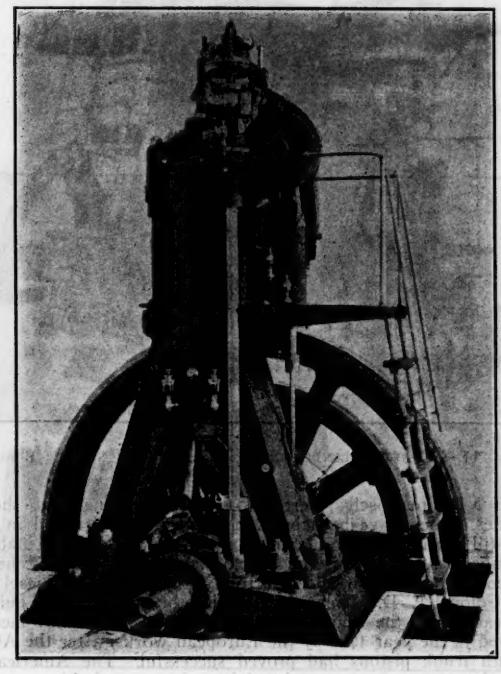


FIG. 10.—TRUNK PISTON TYPE, 70 TO 90 HORSEPOWER, 1901. CYLINDER, 15.75 INCHES IN DIAMETER; 26.62 INCHES STROKE; 150 REVOLUTIONS PER MINUTE.

A three-cylinder engine of the same type, Fig. 11, was made by Sulzer Brothers in 1906. The two latter engines show a slight alteration; the petroleum pump is driven from the vertical instead of from the horizontal cam shaft, as was the case in the previous engines. The engines built by Sulzer Brothers and by Carels have also a rotating stuffing box for the fuel needle. This arrangement was first built in Sweden on the author's instructions, and worked successfully. The well-known 500 horse-power three-cylinder engine of Carels was exhibited at Liége in 1905.

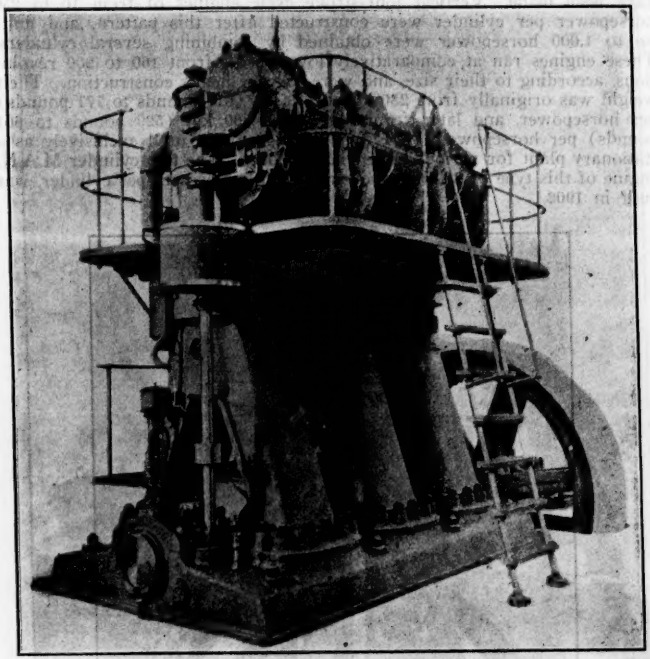


FIG. 11.—THREE-CYLINDER 300-HORSEPOWER ENGINE, 1906.

The author has purposely referred to this type of engine to show that these engines, which have been built in various factories and in various countries, still remain almost an exact copy of the old experimental engine, Fig. 8. Only in America was the design simplified, or rather cheapened, from the commencement by the director of the American Diesel Engine Company, Colonel E. D. Meier. In America the engines were built without crossheads from the beginning, an idea which, as already mentioned, was followed in the year 1901 by the European works, after the American engines with trunk pistons had proved successful. The Americans also built from the commencement a closed base frame, and this construction, as will be seen later, has also been recently adopted in the European high-speed engines, but in a more refined and better form. Moreover, the American engines had no valves in the cylinder covers, Fig. 12, but they

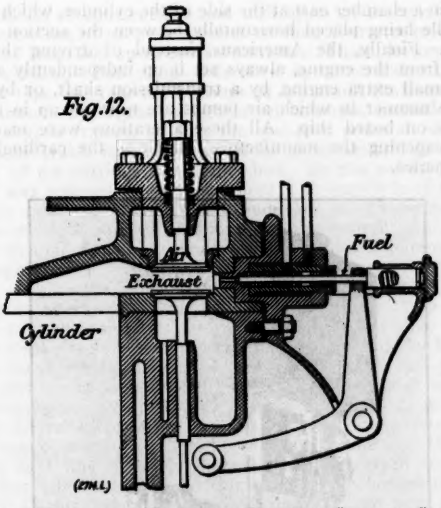


FIG. 12.—VALVES IN CHAMBER CAST ON SIDE OF CYLINDER.

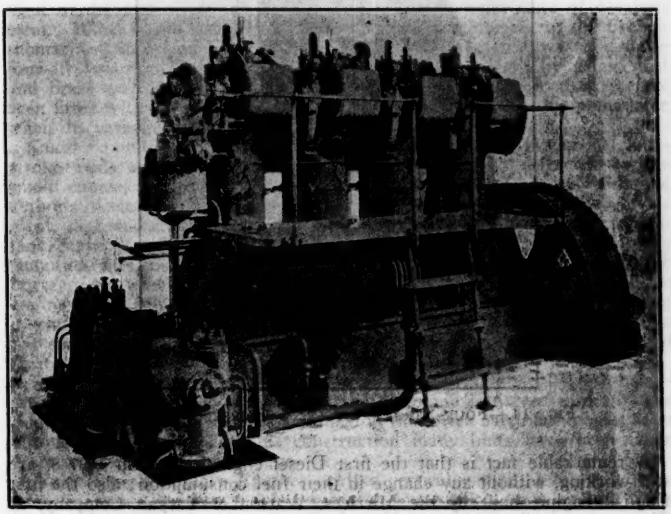


FIG. 13.—FOUR-STROKE CYCLE HIGH-SPEED ENGINE, 1909; SHOWING BOX-FRAME AND AIR PUMP.

were placed in a chamber cast at the side of the cylinder, which necessitated the fuel needle being placed horizontally between the suction and the exhaust valves. Finally, the Americans, instead of driving the air-supply pump direct from the engine, always set it up independently and drove it either by a small extra engine, by a transmission shaft, or by an electric motor, in the manner in which air pumps are now set up in many Diesel engine plants on board ship. All these alterations were made with the object of cheapening the manufacture, which is the cardinal feature in American practice.

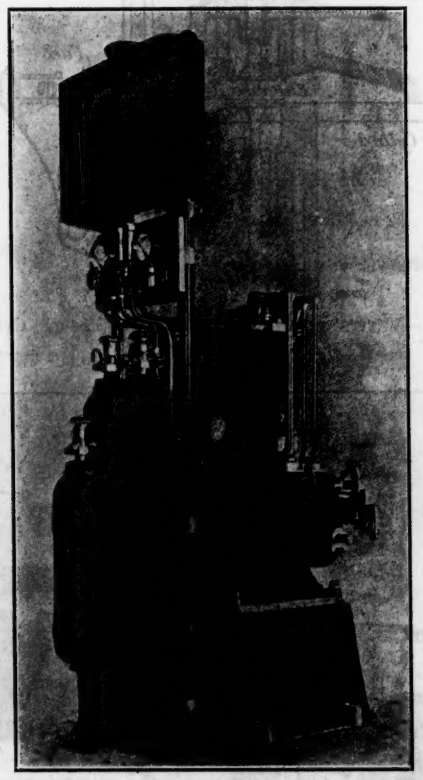


FIG. 14.—FOUR-STROKE CYCLE, 5 HORSEPOWER, 1909.

A remarkable fact is that the first Diesel engines, built in 1897–8, are still working, without any change in their fuel consumption; also the first English engine, built by the Mirrlees, Watson & Yaryan Company, at Glasgow, according to the author's design of fourteen years ago, is still working.

As the central electric stations took up the Diesel engine very early, the necessity for quicker-running engines arose. This need, and the improve-

ment in methods of construction and utilization of materials, caused the gradual introduction of the new quicker-running four-stroke cycle engines, with speeds of from 300 up to 600 revolutions. These, however, were still exclusively vertical. The main difference in construction as compared with the first type was that the bearings of the crank shaft were connected with the cylinders by means of light steel columns, instead of by heavy cast-iron A-shaped frames, so that the cast-iron pedestal of the machine became a light crank case, relieved from great strain; in addition, the thickness of all the castings was diminished. By this means the weight of the engines was reduced to about one-fourth to one-fifth of the weight of the old types, or to about 50 kg. (110 pounds) per horsepower. Engines of this kind are now built up to about 700 horsepower, and are especially suitable for driving dynamos, blowers and centrifugal pumps, and also as auxiliary engines on board large vessels, etc.

The first of these high-speed four-stroke cycle engines made by the M.A.N. had no alterations in the valves, the needle or the gearing, nor in the driving and the position of the petroleum and air-supply pumps, etc.

In a four-stroke cycle high-speed engine, Fig. 13, made by Messrs. Sulzer Brothers in the year 1909, the only difference, except the box-pattern frame, between it and the arrangement of the old type consisted in the position of the air-supply pump, which was in this case fitted to one end of the engine and driven direct from the crank shaft. In a later four-stroke cycle high-speed engine of 350 horsepower, made by Messrs. Sulzer Brothers in 1911, the air-supply pump was also driven from the crank shaft, but was fitted between the cylinders on the box-pattern base in a neater way. In this case also no radical alteration has been made.

These latter kinds of engines may be regarded as the final and permanent type of the vertical four-stroke cycle engine for stationary purposes, both for high and low speeds. With this and similar types the development of the four-stroke cycle engine reached a definite state of development. When in the last decade, through rapid development of the French submarines, an urgent need for a reliable submarine engine was felt, these four-stroke cycle engines were further reduced in weight by using steel and brass castings, with still thinner walls, and they have also recently been fitted with reversing gear. The author will return to this point later when discussing marine engines.

Small Engines.—This summary of the development of the vertical four-stroke cycle engine would not be complete without a reference to the small engines which have recently been built in accordance with the author's designs. Fig. 14 shows a complete 5-horsepower one-cylinder plant, designed in 1909, for 600 revolutions per minute, with petroleum tank, starting and air-admission chambers. The officially recognized consumption for this small engine is 240 grammes (0.53 pound) per brake horsepower, which is therefore not much more than with the old large engines of medium horsepower. At present the author is endeavoring to simplify and strengthen this small engine, which will then be suitable for small manufacturers and for farmers, who are not especially skilled in mechanical work.

Fig. 15 shows a 10-horsepower plant which is composed of two 5-horsepower cylinders of the above kind; the air pump is driven direct by the main shaft. The cam shaft is arranged in the lower part of the base, as in automobile engines. For this construction many hints have been taken from automobile-engine designs.

Horizontal Stationary Engines.—After vertical engines had solely been used for about twelve years, horizontal four-stroke cycle engines were built. The author is uncertain whether this type was a real necessity, or whether it was originally only constructed for purposes of competition to bring out something new; it is, however, not his intention to compare the merits of the two types. The first horizontal engines were practically

only vertical engines laid on their sides, without any independent structural innovations; all the valves were fitted in the cylinder cover, in exactly the same way as was done in the old vertical engine shown in Fig. 9. The valves were actuated by a small cross shaft, which was itself driven by means of gears from another shaft parallel to the cylinder axis. The air pump was fitted in exactly the same way as in the old vertical engine.

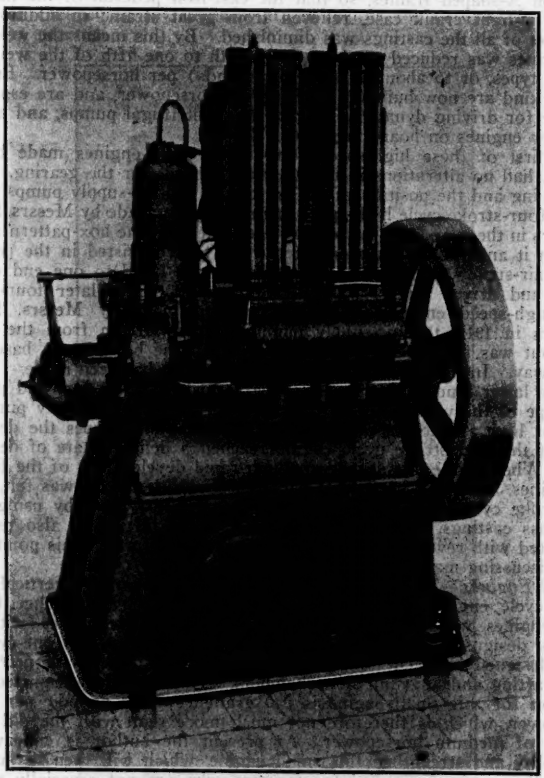


FIG. 15—DOUBLE-CYLINDER FOUR-STROKE CYCLE, 10 HORSEPOWER.

Gradually the designers freed themselves from the tradition of the vertical engine, and some details were altered in such a way that they were more suitable for the horizontal position, and a type of engine was thus obtained which is hardly distinguishable from the horizontal gas engines, as Fig. 16 shows. In this engine, made by the Swiss Locomotive Works, Winterthur, the inlet valves are no longer placed in the cover, but on the side of the cylinder as in gas engines, and are directly driven from the longitudinal cam shaft. A cross cam shaft is no longer used. Only the fuel and exhaust valves are left in the cover, while the air compressor is here arranged in another way.

These designs are today very often used for smaller plants of 20 horsepower and more, especially by gas-engine manufacturers, who took up the construction of Diesel engines on their own account on the expiration of the patents, and who preferred to keep to the old types of horizontal gas engines. But the M.A.N. built such horizontal Diesel engines for very high horsepowers as double-acting four-stroke cycle engines with two or

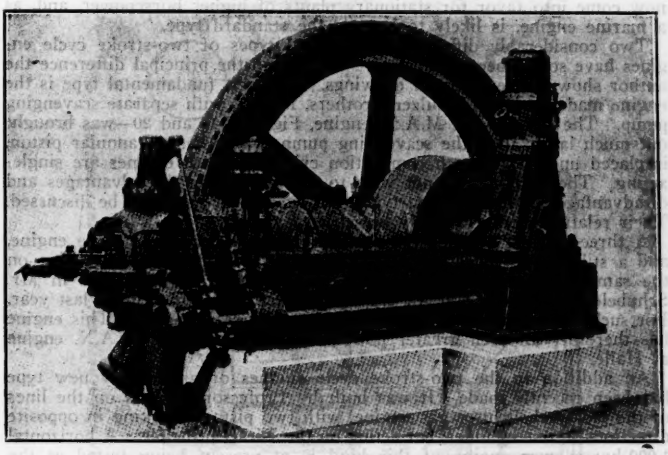


FIG. 16.—HORIZONTAL TYPE, 50 HORSEPOWER.

four cylinders arranged tandem. The largest engine of this kind so far is a double-acting four-stroke cycle tandem twin engine, of 1,600 to 2,000 horsepower, or 400 to 500 horsepower per cylinder, with a speed of 150 revolutions per minute; this engine is working in the Corporation gas works at Halle, using water-gas tar as fuel. It was constructed on the lines of the well-known Nürnberg large gas engine for blast-furnace gas.

TWO-STROKE CYCLE ENGINES.

As very often stated by the author, the Diesel principle is essentially suitable to a two-stroke cycle engine, because the scavenging is not done with a fuel-air mixture, but with pure air, so that not only untimely ignitions, but also fuel losses are avoided, and the scavenging can be done more effectively, and with almost any quantity of air desired.

The first two-stroke cycle engines on the Diesel principle were built in 1900 and 1901, in Germany and England, after drawings made by Güldner, but without success, because these drawings still followed too closely the two-stroke cycle gas engines, and because the constructional arrangements were unsuited to the Diesel engine.

Successful attempts to construct a two-stroke cycle Diesel engine on entirely new lines have, however, been made recently by Messrs. Sulzer Brothers, of Winterthur, so that today this type is on a nearly equal footing with the old four-stroke cycle engine. This has been effected by working entirely on the original Diesel principle. The author says "on a nearly equal footing," because the four-stroke cycle engine still has a

better combustion and a more economical fuel consumption, and is, above all, simpler in its method of working. It thus remains the standard perfect engine, and still predominates for medium-sized stationary plants up to 500 or 600 horsepower (no exact limit can be given) wherever the highest perfection and the greatest economy are desired; but engineers are now doubtful whether this supremacy will last much longer. On the other hand, the two-stroke cycle engine, with its smaller cylinders, has now come into favor for stationary plants of higher horsepower, and, as a marine engine, is likely to become the standard type.

Two considerably different fundamental types of two-stroke cycle engines have so far been competing. To explain the principal difference the author shows some sectional drawings. The first fundamental type is the engine made by Messrs. Sulzer Brothers, Fig. 17, with separate scavenging pump. The second—the M.A.N. engine, Figs. 18, 19 and 20—was brought out much later; in it the scavenging pump, which has an annular piston, is placed underneath each combustion cylinder. Both engines are single-acting. The author does not wish to comment upon the advantages and disadvantages of the different types, but would leave them to be discussed. Their relative merits can only be settled by experience.

A three-cylinder 750-horsepower Sulzer-Diesel two-stroke cycle engine, and a still larger Sulzer-Diesel four-cylinder two-stroke cycle engine on the same system, of 2,000 to 2,400 horsepower, were illustrated in Mr. Schubeler's paper, read at the Zürich meeting of the Institution last year. For such large engines two scavenging pumps are necessary. This engine has therefore an even greater power than the horizontal M.A.N. engine at Halle.

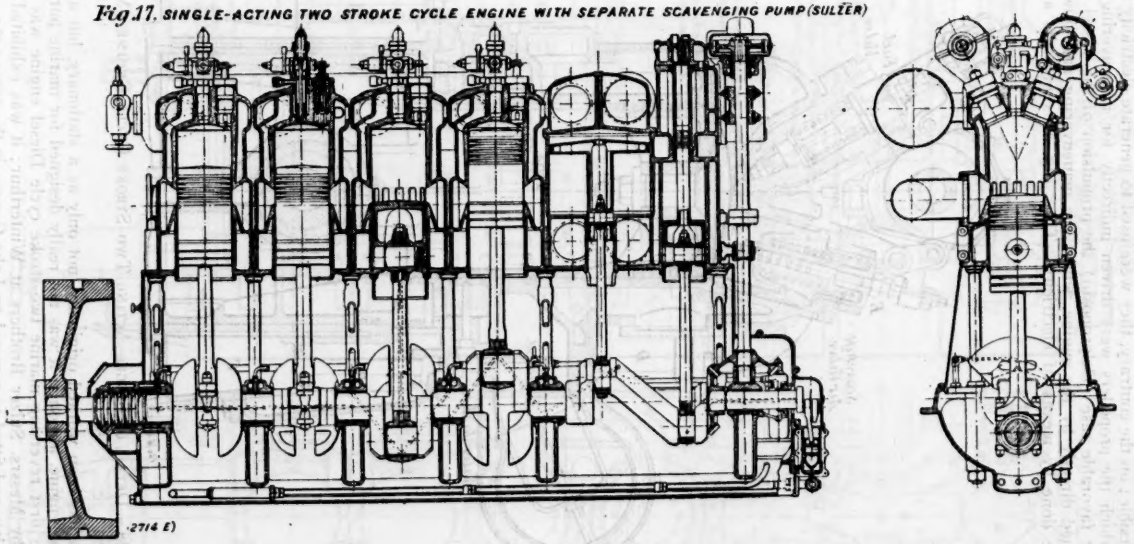
An addition to the two-stroke cycle engines of an entirely new type has been recently made. It was built by Professor Junkers on the lines of the old Oechelhäuser gas engine, with two pistons working in opposite directions in one cylinder, but acting on the Diesel principle. A horizontal 1,000-horsepower engine of this kind is at present being tested at the laboratory of Professor Junkers at Aix-la-Chapelle. The merits of this type of engine are left for discussion.

Marine Engines.

The first marine Diesel engine of 20-horsepower, Fig. 21, was constructed in 1902–3, in France, for use on a canal boat, by the French engineers Adrien Bochet and Frédéric Dyckoff, in conjunction with the author. This engine had, like the Junkers engines already mentioned, two pistons working in opposite directions in one cylinder, but the flywheel shaft was not at one end of the cylinder as in the Junkers engine, but traversed a cooled chamber passing straight through the combustion chamber. The engine worked on a four-stroke cycle. The great feature of this arrangement was the very high speed which was made possible by the perfect balance. This small engine was, as stated, used to drive a canal boat and worked quite satisfactorily. Others were also built in various sizes up to several hundreds of horsepower for some French submarines by Sautter, Harlé & Co., Paris.

This type of engine is of no further practical interest today; but while its first application to a canal boat is of no importance in itself, it has at least the historical interest of being the first Diesel engine to be used on a boat. Since the date named the evolution of the Diesel marine engine has steadily continued, chiefly on the demand of the French submarines and Russian river boats. The author has already mentioned that later on the high-speed four-stroke cycle engines, built for electric-power stations, were made even lighter than before, and used for French submarines and for Russian river vessels. These engines were not originally

Fig 17. SINGLE-ACTING TWO STROKE CYCLE ENGINE WITH SEPARATE SCAVENGING PUMP (SULZER)



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reversible; on the contrary, they were used to generate electricity by means of which the propellers were driven indirectly for maneuvering. In the most favorable case (Delproposto) the propulsion of the vessel was performed directly by the engine, whilst the maneuvering and slow driving were done by means of electricity. Thus Fig. 13 represents an engine

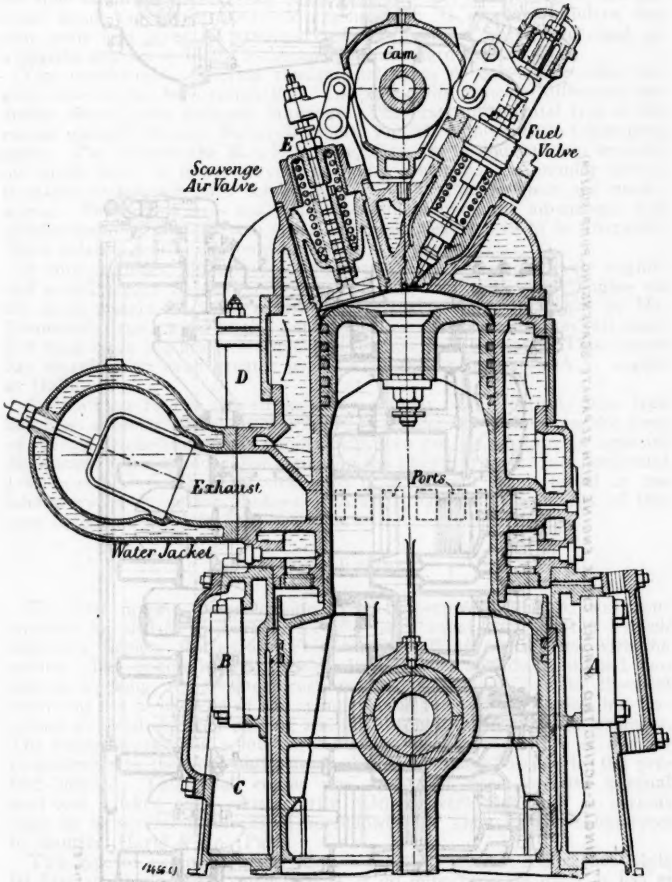
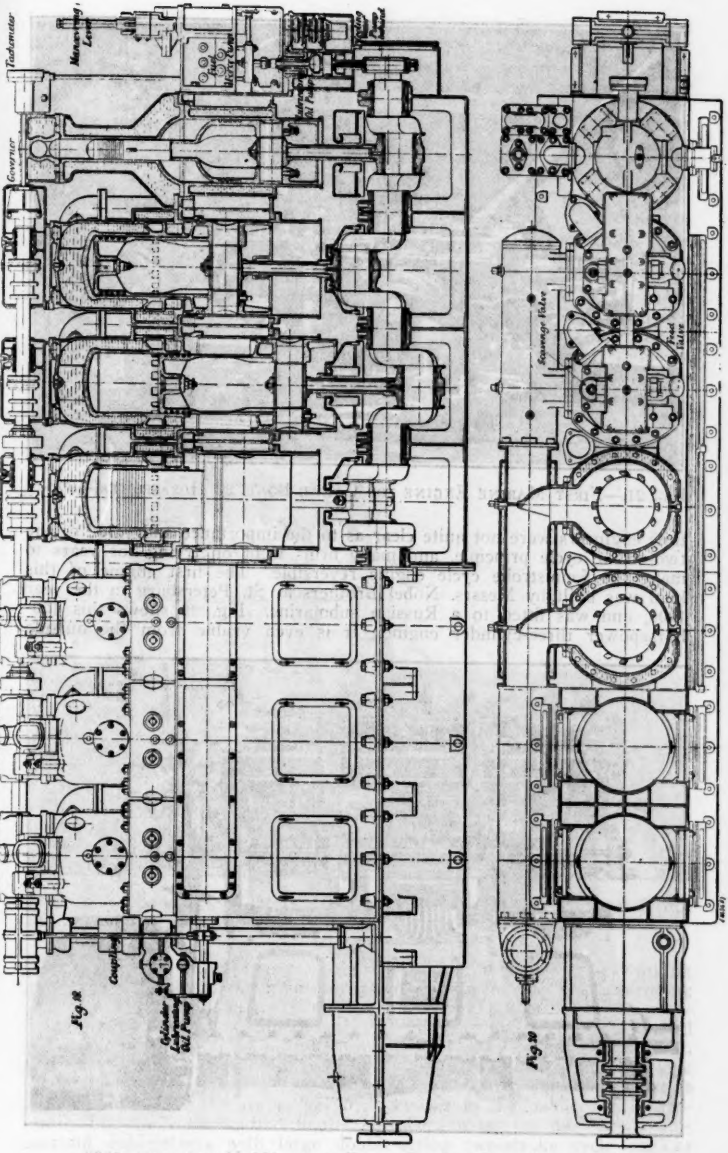


FIG. 18.—DETAIL OF SINGLE-ACTING TWO-STROKE CYCLE ENGINE (M.A.N.).

which has been worked originally, not only as a stationary, but also as a marine engine, although it was not really designed for marine purposes.

The first reversing marine two-stroke cycle Diesel engine was built in 1905 by Messrs. Sulzer Brothers at Winterthur; it was exhibited in 1906 at Milan, and fitted to a vessel on Lake Geneva in the same year. At that


Tachometer
Governor
Coupling
Scavenger Valve
Fig 49
Fig 50

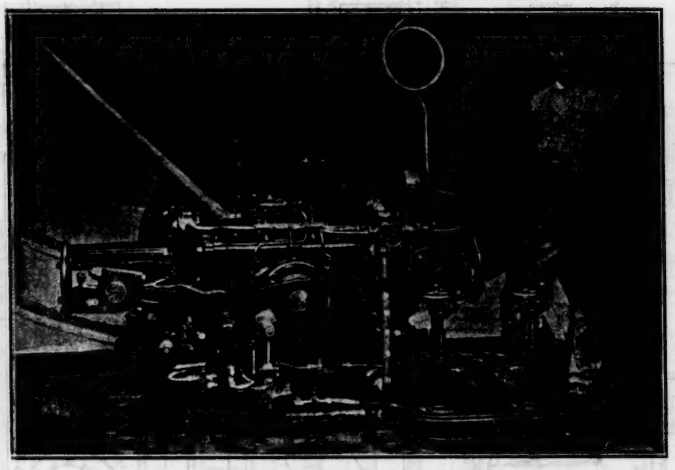


FIG. 21.—FIRST MARINE ENGINE FOR CANAL BOAT, 20 HORSEPOWER, 1902-3.

time engineers were not quite clear as to the importance and value of the two-stroke cycle principle, and many firms went on trying for years to make the four-stroke cycle engine reversible. The first engine of this kind was built by Messrs. Nobel Brothers at St. Petersburg in the year 1908, and was fitted to a Russian submarine. Fig. 22 shows this 120-horsepower three-cylinder engine. It is even visible from the outside

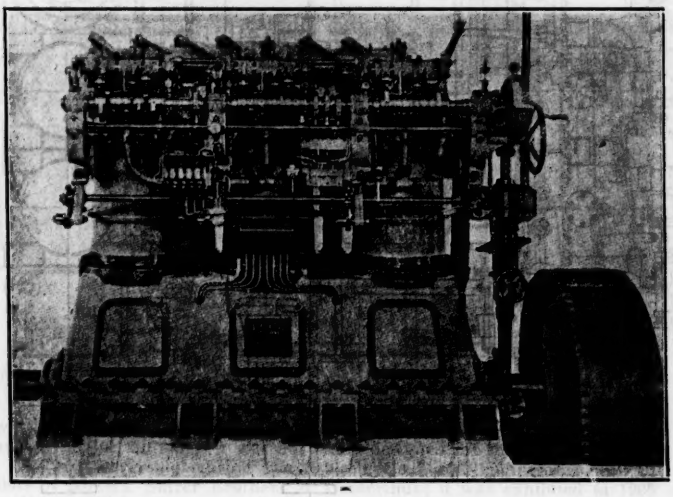


FIG. 22.—FOUR-STROKE CYCLE ENGINE, 120 HORSEPOWER, 1908.

view what great mechanical complications were at first caused by the reversing of the four-stroke cycle engine. This problem has recently been solved in a much more simple and neater way, in a six-cylinder 150-horsepower reversible four-stroke cycle engine of 350 revolutions, constructed in the year 1911 by the French firm, Messrs. Delaunay-Belleville. This engine is fitted with two air pumps, of which a spare one is for maneuvering. In many factories reversible four-stroke cycle marine engines are still built, but, on the whole, engineers are, for navigation purposes, inclined to abandon the four-stroke cycle engine entirely and to replace it by the two-stroke cycle engine.

The small four-cylinder engine of 30 horsepower and 600 revolutions per minute, illustrated in Fig. 23, is also a reversible four-stroke cycle engine. It was built for experimental purposes in the year 1909, after designs by the author, as an automobile engine for heavy loads, but it can also easily work as a marine engine. The cam shaft is mounted on the cylinder cover, and the illustration shows the engine with the cover lifted. The illustration above referred to is again of historical value in so far as it illustrates the first attempt to construct the Diesel engine as an automobile engine for traction wagons, and no doubt in future years these experiments, carried out in some different way, will lead to satisfactory results.

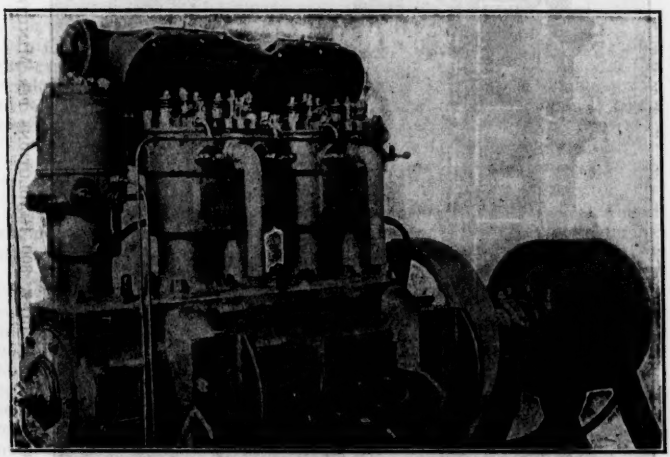


FIG. 23.—EXPERIMENTAL REVERSIBLE FOUR-CYLINDER FOUR-STROKE CYCLE ENGINE, 30 HORSEPOWER, 600 REVOLUTIONS PER MINUTE, 1909.

Fig. 24 shows a quite new Sulzer six-cylinder marine engine working at 300 revolutions, in which an innovation may be noted. The scavenging valves are not fitted on the top within the cover, but below in the scavenging air reservoir. Quite recently very large cylinder units single and double-acting vertical two-stroke cycle Diesel engines have been built. Fig. 25 shows one of these single-acting two-stroke cycle engines with one cylinder of 1,200 horsepower, made by Messrs. Carels Frères, the results of the tests of which are as yet little known to the public. It is generally known, however, that in the Nürnberg works of the M.A.N. important experiments with large double-acting two-stroke cycle engines are being carried out.

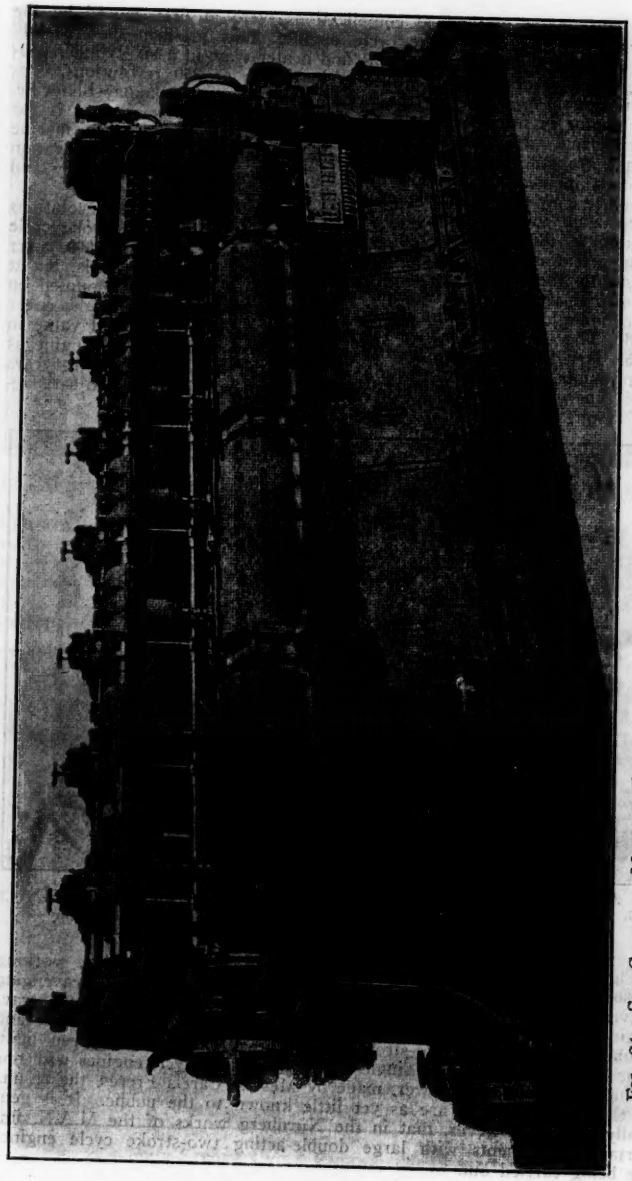


FIG. 24.—SIX-CYLINDER MARINE ENGINE, 300 REVOLUTIONS PER MINUTE, 1911; SCAVENGING VALVES BELOW.

FIG. 25.—SINGLE-CYLINDER SINGLE-ACTING TWO-STROKE CYCLE ENGINE. 1,200 HORSEPOWER, 1911.

In these works prolonged official tests of a three-cylinder double-acting two-cycle engine of 850 horsepower were made in August, 1911, and the results proved highly satisfactory. At the present time a double-acting two-cycle three-cylinder engine of 2,000 horsepower per cylinder is being tested at Nürnberg. The dimensions of these cylinders are as follows:

Diameter	800 mm. (31½ inches).
Stroke	1,060 mm. (41¾ inches).
Revolutions per minute.....	160

The air-supply pump for fuel injection is driven by a special Diesel engine, whilst the scavenging pumps are driven direct from the crank shaft.

If, as seems probable, these tests also give satisfactory results, the era of very large Diesel engines has begun, especially low-speed marine engines suitable for driving propellers. This last engine has already yielded considerably more than 2,500 horsepower per cylinder, so that an engine unit of this kind with six cylinders would give 15,000 horsepower, or 45,000 horsepower for a vessel with three propellers. This kind of marine engine requires six cylinders to ensure a regular turning moment and balancing, so that the number of cylinders cannot be considered abnormally high; on the contrary, it must be accepted as the most suitable and proper number. The opinion which was still prevalent some months ago, that about 1,000 horsepower per cylinder is the maximum for Diesel engines, has therefore been quickly overthrown by these facts, and it may be safely assumed that the day of the large marine engine is already very near at hand. At Messrs. Krupp's Germania Works cylinder units of 2,000 horsepower double-acting two-cycle are being tested at present; also at Sulzer's works a single-acting two-cycle cylinder of 2,000 horsepower, and at Messrs. Vickers' works a three-cylinder single-acting two-cycle, of 2,000 horsepower per cylinder, are being constructed.

Junkers' system is used for vertical marine engines. The author regrets that he has not a photograph of a finished engine.

From motives of prudence the various navies which are now fitting some large warships with Diesel engines started with one Diesel only out of the two or three engines on board; the Diesel works alone when the ship is running at normal speed, but for high speed steam is used as an auxiliary. It is evident that larger warships will not be fitted solely with

Diesel engines until practical tests on the high seas have proved to be completely successful.

Conclusions.—As will be seen from this historical summary, countless different types of Diesel engines have been so developed that it has become very difficult for even the expert to choose between them. If one looks through the technical press, numerous other schemes will be found on which the author will not dwell, as they have never been actually carried out. Today, if a new firm is about to manufacture Diesel engines, it is almost impossible to give them sound advice as to the type and size they should choose from this bewildering variety. The fads, habits and tastes of the purchasers, and the kinds of machine tools in use in the works, have to be taken into consideration rather than technical points of the engine itself. Development is proceeding so rapidly at present that, within a few months, opinions even on important points may easily be changed.

Still, the author believes that this period of chaotic production will soon be over. At present it is generally agreed that the four-stroke cycle engine from 5 up to 600 horsepower may be regarded as the exclusive type for stationary plants; but it will probably not remain so much longer in spite of its perfection, in view of the development of the two-stroke cycle engine, especially that of the double-acting type.

The use of the two-stroke cycle engine for stationary work has increased, but it is anticipated that this will be still further extended. Although this engine may never equal the four-stroke cycle engine as regards thermal efficiency, its initial cost is so much lower that its slightly higher fuel consumption will be more than counterbalanced by the greater interest and amortisation on the higher-priced four-stroke cycle engine. When this stage is reached, the question is simply one of economy. In the author's opinion the two-stroke cycle engine will thus soon make headway for stationary plants. It will therefore be necessary to produce from the various systems of two-stroke cycle engines a simple standard type, with which the more complicated types will not be able to compete. It is the author's belief that this simple standard type will make its appearance very soon, and that thus the Diesel engine movement will leave the unsettled stage and enter on a period of quiet expansion.

In Appendix IV is given a list of vessels propelled by Diesel engines, which is as complete as the author has been able to make it from private information and from publications. As many firms do not publish the details of their work, and as very little information can be obtained about anything connected with warships, such a list cannot claim to be complete. It gives only a general idea of the immense amount of work which is being done at present in this direction.

In Appendix V some brief results of vessels propelled by Diesel engines are given, as far as it has been possible to obtain them.

Special Importance of the Diesel Engine for Great Britain.

After this short summary about the importance of the Diesel engine for the world's industry in general, and the historical review of Diesel types, the author desires to add a few words on the importance of the Diesel engine especially for Great Britain. In this connection the three following facts must be borne in mind:

(1) Great Britain is an exclusively coal-producing country.

(2) Great Britain has the largest Colonial Empire in the world; and

(3) Great Britain is the greatest shipping nation in the world.

Dealing with (1), Great Britain has had, at least until today, no natural liquid fuels of its own; it is an exclusively coal-producing country. Based on these statements, it has often been seriously put forward in recent times that England has no interest in the Diesel engine, and that it is against her most vital interest to participate in the development of this engine, as she would neglect her wealth in coal, and make herself dependent on

foreign markets when using liquid fuels. The Exhibition at Turin has shown clearly that England has still very little interest in the Diesel engine as far as stationary plants are concerned, owing to her abundance of coal; for in the English Section, with the exception of some very small oil engines, only generator-gas engines were exhibited; but these were numerous, of first-class workmanship, and were built by the best firms in the country. England has therefore made great progress in power generating with gas, while she has up to the present given relatively little attention to power generating with liquid fuels.

But the foregoing arguments in favor of coal are not correct; the contrary is true. England has the greatest interest in replacing the coal-wasting steam engine by the more economical Diesel engine, and this, firstly, because she can therewith effect enormous savings in her most valuable treasure—coal—and thus defer the exhaustion of her stock; and, secondly, because she can run her coal industry and the independent chemical industries on more economical lines when using the coal in the more rational and refined way, as already mentioned. Finally, because she will also make herself free and independent of foreign markets for the supply of liquid fuels by using coal in this way—that is, by working the tars and tar oils in the Diesel engine. It is not intended to imply that the whole demand of fuel for England could be produced in this way in the country itself, but through the inland production, increase of price for foreign oils and the establishment of trusts and monopoly companies will be prevented. In this sense independence is meant.*

(2) *Great Britain has the Largest Colonial Empire in the World.*—It is not possible to foresee today what England can obtain for her Colonies from the Diesel engine; even when using only the natural mineral oils, the Diesel engine is the predestined colonial engine, because only about the fourth or sixth part of weight in fuel has to be transported for it to the Colonies and their hinterlands, as compared with the steam engine. For a colonial engine the cost of freightage for fuel is generally the determining factor. Further, the transport of these liquid fuels is considerably easier and more convenient than the transport of coal, especially when tank vessels and pipe lines are used; and, finally, the difficulty of working a boiler plant will only occasionally come into consideration in the Colonies, especially in the interior, except, of course, for small plants using wood, straw and the like.

It may be mentioned, in this connection, that a pipe line of 400 km. (about 250 miles) in length for crude petroleum is being built on the River Congo from Matadi to Leopoldville, by means of which this extensive district will be supplied, in the simplest and cheapest way, with a constant flowing fuel source, from which navigation and the railways, agriculture as well as other industries, will get their element of life—namely, motor power. This magnificent example ought to be followed in the English colonies; it is not necessary to specify in detail the great importance of such an undertaking for them.

If one also considers that the Diesel engine can utilize vegetable oils, entirely new prospects are brought to light for the cultivation and expansion of industry in the colonies, which are for no other country of such eminent importance as for Great Britain; and this is where she ought to start as soon as possible. The Diesel engine can be worked with the colonies' own resources, and thus again can influence to a great extent

* In this connection it may be of some interest to state that the tar production of Germany is sufficient for more than 5 milliards of horsepower-hours per year, which means about 1¼ millions of horsepower running 300 days of ten hours each all the year. In case of war and cutting off the supply of foreign fuel, this quantity would be entirely sufficient for running the whole fleet, war and mercantile, and for providing in the meantime the power for the inland industry as far as necessary. The author has not the figures for England, but he presumes they are of similar significance.

the further expansion of agriculture in districts where it is predominant. This sounds today somewhat like a dream, but the author ventures to prophesy, with full conviction, that this way of using the Diesel engine will one day be of the utmost importance.

3. Finally, *Great Britain is the Greatest Shipping Nation in the World.*—When the first successes of the Diesel engine, as a marine engine, were heard of in England recently, and it was published that already numerous small mercantile and war vessels were fitted with Diesel engines, and the possibility of more important plants was mentioned, and when it was realized that larger ocean vessels were destined to be fitted with Diesel engines, and that even a warship with a very large Diesel engine was in construction, it provoked a great movement and excitement in Great Britain which is still fresh in our minds. Moreover, the reports about successful voyages with Diesel vessels under very difficult weather conditions are already increasing in number. The captains who have had Diesel engines on their vessels report on the great security and comfort in working; shipbuilders publish the figures of their savings. It is unquestionable that one of the greatets evolutions of modern industry will be connected with this development of the Diesel engine, and that Great Britain, as the greatest shipping nation of the world, will derive the greatest advantage from it.

Appendix I.

Suitable Oils.

The Swiss Fuel-Testing Laboratory at the University of Zurich, under the direction of Professor Constam, decided to undertake the examination of the qualities and composition of all liquid fuels which can be used for Diesel engines. These researches included the investigation of the following points:

1. On the physical properties, such as
   (*a*) Properties when cold,
   (*b*) Properties on heating (boiling analysis).
2. Chemical properties, such as:
   (*a*) Chemical constituents.
   (*b*) Percentage of H_2O and ash.
   (*c*) Calorific power.

This laboratory is destined to become a center for the investigation of fuels for Diesel engines on account of the perfection of its equipment, the accuracy of its work, and the excellence of its management. The Laboratory intends to publish from time to time exhaustive reports of its researches.

From tests and examinations already made, power oils have been divided into the following three classes:

1. *Normal Oils which can always be Used:*

(*a*) Mineral oils freed from benzine (gas-oils).	Hydrogen over 10 per cent. Calorific power, per kilogram, over 10,000 calories (39,680 B.T.U.). No solid impurities.
(*b*) Lignite tar oils.....	Hydrogen over 10 per cent. Calorific power, per kilogram, over 9,700 calories (38,489 B.T.U.).
(*c*) Fat oils from vegetable or animal sources, such as earth-nut oil, castor-oil, fish oils, etc.*	Scarcely any researches have been made on these. Earth-nut oil has 11.8 per cent. hydrogen, and calorific power, per kilogram, 8,600 calories (34,124 B.T.U.).

Tars from horizontal or inclined retorts.

* This class of oil has been added by the author from his own investigations of earth-nut oil.

2. *Oils which can be used only with the aid of special apparatus:*
   (*a*) Pit coal-tar oil.
   (*b*) Vertical-oven, water-gas, and oil-gas tars, probably also coke-oven tars, the tests on which have not yet been completed.

General characteristics:
   Hydrogen not over 3 per cent.
   Amount of free carbon not over 3 per cent.
   Residue on coking not over 3 per cent.
   Calorific power not under 8,600 calories per kilogram (34,124 B.T.U.).

3. *Oils which cannot be used:*
   Tars from horizontal or inclined retorts.

It must not be understood that these will not be used in Diesel engines under special conditions; but, on the whole, the above classification is accurate in the present state of development of the Diesel engine.

It is evident that for estimating the value of power oils, not only the above qualities, but all their chemical and physical properties must be considered, which is only possible after a thorough investigation of each kind of oil.

Appendix II.

Specifications of Tar-Oil Suitable for Diesel Engines.

(*From the German Tar-Production Trust at Essen-Ruhr.*)

1. Tar oils should not contain more than a trace of constituents insoluble in xylol. The test on this is performed as follows:—25 grammes (0.88 oz. av.) of oil are mixed with 25 cub. cm. (1.525 cub. in.) of xylol, shaken and filtered. The filter-paper, before being used, is dried and weighed, and after filtration has taken place it is thoroughly washed with hot xylol. After re-drying the weight should not be increased by more than 0.1 gramme.

2. The water contents should not exceed 1 per cent. The testing of the water contents is made by the well-known xylol method.

3. The residue of the coke should not exceed 3 per cent.

4. When performing the boiling analysis, at least 60 per cent. by volume of the oil should be distilled on heating up to 300 degrees C. The boiling and analysis should be carried out according to the rules laid down by the Trust.

5. The minimum calorific power must not be less than 8,800 calories per kilogramme. For oils of less calorific power, the purchaser has the right of deducting 2 per cent. of the net price of the delivered oil for each 100 calories below this minimum.

6. The flash point, as determined in an open crucible by Von Holde's method for lubricating oils, must not be below 65 degrees C.

7. The oil must be quite fluid at 15 degrees C. The purchaser has not the right to reject oils on the ground that emulsions appear after five minutes' stirring when the oil is cooled to 8 degrees.

Purchasers should be urged to fit their oil-storing tanks and oil pipes with warming arrangements to re-dissolve emulsions caused by the temperature falling below 15 degrees C.

8. If emulsions have been caused by the cooling of the oils in the tank during transport, the purchaser must re-dissolve them by means of this apparatus.

Insoluble residues may be deducted from the weight of oil supplied.

APPENDIX III.—PROPERTIES OF PIT-COAL TARS.

(*By* W. ALLNER. *On the Application of Tar for the Working of Diesel Engines.*)

	Specific gravity.	Water.	Free carbon.	Flash point.	Burning point.	Viscosity at degree C.						Thermal units (higher calorific power) per 1 kg. (undried tar).	Analysis of dry tar.			Percentage.
						20	50	70	75	80	100		C.	H.	S.	
		p. c.	p. c.	deg. C.	deg. C.											
Vertical-oven tar (gas works of the D.C.G.G.):																
English coal	1.124	5.0	3.65	44	100	39.3	3.9	2.2	...	1.7	...	...	...	...	...	...
Do	...	3.0	1.87	71	...	75.5	4.5	2.2	...	...	...	8,986	...	...	...	...
Do.	...	2.1	1.25	45	104	7.8	2.5	1.5	...	...	...	9,185	...	...	...	...
Westphalian coal	...	2.0	1.17	42	99	10 1	2.1	1.4	...	...	...	8,890	...	...	...	...
Upper Silesian coal	...	1.0	0.23	40	75	6.9	1.8	1.3	...	...	...	9,132	...	...	...	...
Westphalian coal	1.123	1.0	1.66	...	...	...	...	...	...	...	...	9,103	...	...	1.42	...
Do.	1.182	1.5	1.60	...	...	...	...	...	...	...	...	9,015	...	...	0.50	...
Westphalian and English coal	1.091	1.7	4.26	...	...	...	...	...	...	...	...	9,163	...	...	0.33	...
English and Westphalian coal	1.092	2.9	1.13	...	...	...	...	...	...	...	...	9,129	...	...	0.52	...
Silesian coal	1.097	1.0	1.86	...	...	...	...	...	...	...	...	9,378	...	...	0.48	...
Horizontal-oven tar (gas works of the D.C.G.G.):																
English coal	...	4.0	21.7	71	125	114.4	16.0	7.2	...	...	...	8,766	...	...	...	...
Do.	1.246	8.0	28.8	90	130	550.0	51.0	23.0	...	18.9	...	...	...	...	...	...
English and a little Silesian coal	1.231	5.5	32.99	...	...	...	...	...	...	...	...	8,615	...	...	1.25	...
Do.	1.160	6.6	14.85	...	...	...	...	...	...	...	...	8,576	...	...	0.46	...
Silesian coal	1.210	4.9	23.76	...	...	...	...	...	...	...	...	8,709	...	...	0.35	...
Westphalian coal	1.228	3.6	26.79	...	...	...	...	...	...	...	...	8,729	...	...	0.46	...
Vertical-oven tar (Cologne Gas Works):																Pitch
Westphalian coal	1.12	...	1.5 ÷ 2.5	...	...	...	...	...	...	...	...	...	...	...	...	25–30
Horizontal-oven tar (Cologne Gas Works):																
Westphalian coal	1.20	...	20 ÷ 25	...	...	...	...	...	...	...	...	...	...	...	...	55–50
Vertical-oven tar (Zurich Gas Works):																
Ruhr coal (one-third less water than coke-oven tar when stored for an equal period)	...	...	ca. 5	...	...	...	...	...	...	...	...	...	...	...	...	...
Coke-oven tar (Zurich Gas Works—older ovens):																
Ruhr coal	...	...	15 ÷ 25	...	...	...	...	...	...	...	...	...	...	...	...	...
Horizontal-oven tar (various kinds of coal):																
Saar coal	...	7 9	31.7	...	...	...	38.0	...	24.6	...	8 3	8,910	92.48	4.44	0.25	...
Ruhr coal	...	9.3	35.2	...	...	...	...	...	43.1	...	...	8,855	92.48	4.29	0.17	...
Silesian coal	...	11.0	29.2	...	...	...	...	...	19.4	...	...	8,835	92.37	4.59	0.25	...
English coal	...	9.6	36 9	...	...	...	...	...	...	...	132.0	8,826	92.8	4.21	0.42	...
For comparison:																
Paraffin oil	0.893	...	...	81	113	1.36	1.11	1.02	...	...	...	10,511	...	...	...	...
Tar oil	1.008	...	...	63	97	1.38	1.15	1.04	...	...	...	9,470	...	..	...	...

Appendix IV.

List of Vessels Propelled by Diesel Engines.

(Built or in course of construction up to November, 1911; and compiled from publications and private sources.)

As a good many firms maintain strict secrecy about their work in this direction, and as the information obtainable about naval ships is both scarce and unreliable, the lists are not complete. In any case the figures are minima.

(*a*) *Oil-Tank Vessels* (21).—Of these there are:—

Twelve Russian: On the Volga and the Caspian Sea, principally transporting oil for Messrs. Nobel Brothers, St. Petersburg. The largest of these vessels is the *Djelo*, 5,000 tons displacement, 1,000 to 1,200 horsepower.

Four German: For the German-American Oil Company (Standard Oil Company). Largest vessel, 15,000 tons displacement, 3,500 to 4,200 horsepower. In construction at Messrs. Krupp's Germania Works. The vessel will be constructed on the Isherwood system, and is to contain 22 tanks.

Three Dutch: For different oil companies, one of which is the Anglo-Saxon Petrol Company, London. The largest vessel is of 9,300 tons displacement and 2,200 horsepower.

One English: For Messrs. Furness, Withy & Co., Hartlepool. The ship, 3,150-ton cargo boat, is being built by Sir Raylton Dixon & Co., of Middlesbrough, and the engines—800 brake horsepower Carels-Westgarth two-stroke engines, with four cranks—are by Messrs. Richardsons, Westgarth & Co., of Middlesbrough.

One American: For Canadian Lakes, for the Standard Oil Company, New York.

It is known that Messrs. Nobel, of St. Petersburg, are gradually converting their whole fleet on the Volga and Caspian Sea, consisting of fifty to sixty ships of different sizes, into Diesel-engined vessels. It is not known how many of them have already been thus converted.

(*b*) *Tugs* (37).—Of these, most (about twenty-two) are Russian, for use on the Caspian Sea and the Russian tributaries.

Two of them have a towing capacity of 13,000 kg. (12.8 tons) with 800 horsepower; three of them have a towing capacity of 8,200 kg. (8 tons) with 600 horsepower. The power of the others averages about 400 horsepower.

Three German: On the River Elbe (150 horsepower); on the River Weser (200 horsepower); on the River Rhine (400 horsepower).

The remainder are for various countries—namely, Chili, Italy, Sweden, Roumania, etc.

(*c*) *Motor Sailing Vessels* (8).—Of these, two French boats are the most noteworthy:—

French: 1. The four-master *Quevilly*, with about 600 tons displacement, and 600 to 700 horsepower. 2. The five-master *La France*, the largest sailing vessel in the world, with about 11,000 tons displacement and 1,800 to 2,000 horsepower.

One German, one English, one Dutch, one Swedish, and one Italian motor sailing vessels.

(*d*) *Merchant Vessels* (*about* 43) (*Freight, Passenger, and Combined Freight and Passenger Vessels*).—Of these:—

Nine are Russian, including six passenger vessels on the Volga; about 2,000 tons displacement and 1,200 horsepower.

Seven German, including:—One vessel, built by Blohm & Voss for the Hamburg-America Line, of 9,000 tons displacement and 3,000 horsepower; one vessel for the Hamburg South America Steamship Company, of 6,000 tons displacement and 850 horsepower.

Seven Scandinavian, including three vessels for the Danish East Asiatic Company, of 2,500 to 3,000 horsepower.

Four Dutch, three English, two Swiss, two Congo Free State, and the others various, including:—One freight and passenger vessel *Romagna*, on the Adriatic Sea, for the Ravenna-Trieste-Fiume service, of 1,000 tons displacement and 800 horsepower. This vessel, unfortunately, sank a few months ago, owing to faulty loading of its cargo.

Most of these vessels, especially the larger ones, will not be put in service until 1912. Of the Russian passenger vessels, a good many are already in operation.

(*e*) *Fishing-Boats* (14).—Of these:—

Eleven are German herring boats, of 254 tons displacement and 90 horsepower, for the North Sea.

Two English whalers for the South Polar Sea, with 200 horsepower, etc.

(*f*) *Miscellaneous* (*about* 16).—Of these:—

Three are Russian, including:—One private yacht of 200 horsepower, and two for the Russian Minister of the Interior.

Three French:—Private yachts, including one of about 400 tons displacement and 400 to 500 horsepower.

Two German:—The lightship *Hamburg* and the test ship *Bremen*.

One Norwegian, *Fram*, exploring vessel for the South Pole, Captain Amundsen; 380 tons capacity and 180 horsepower.

(*g*) *Submarine* (*about* 110 to 115), including, as far as can be ascertained, up to the end of 1910:—

About 60 to 70 French, of 390 to 800 tons displacement and 600 to 5,000 horsepower; about thirteen Russian, thirteen English, two Dutch, two Danish, one Norwegian, one Swedish, one American, of 1,200 horsepower, two Italian, and two Austrian.

It is probable that the number of vessels has been considerably increased in 1911, and that a large number will be added in the future. Various countries are not mentioned in the list; for instance, Germany, Japan, the South American States, etc. It is almost impossible to publish a list which is absolutely accurate, owing to the secrecy about the number of vessels in construction and the kinds of engines used on them.

(*h*) *Warships* (*about* 20 to 30):—

Here, again, it is impossible to give accurate figures, as the information published is very scarce and contradictory. It is, however, known that several navies are experimenting with Diesel engines on a large scale. The following are especially noteworthy:—A great number of gunboats and a few mine-laying boats for the Russian Navy. Also one French, one English torpedo-boat destroyer, and one Italian.

(*i*) *Small Boats* (*about* 15 to 20):—

In the list of navy vessels, although they are not strictly men-of-war, must be included the small boats such as pinnaces and special boats, of which about fifteen to twenty are destined for the German, Swedish, English, Argentine and Russian Navies.

Summary.

(*a*) Oil-tank vessels, about	21
(*b*) Tugs, about	37
(*c*) Motor sailing vessels, about	8
(*d*) Merchant vessels, freight, passenger, and combined vessels, about.	43
(*e*) Fishing boats about	14
(*f*) Miscellaneous, about	16

(*g*) Submarines, about	115
(*h*) Warships, about	30
(*i*) Small boats, about	20
Total	304

Appendix V.

Brief Results from some Vessels Propelled by Diesel Engines (see Appendix IV).

(*a*) Oil-Tank Vessels.

Russian Oil-Tank Vessel "Vandal":—

Put in commission in 1904. 1,150 tons displacement, 360-450 horsepower. The first oil-tank vessel built for the Volga and Caspian Sea. Reversing (electric) from full-speed ahead to full speed astern in 8 to 10 seconds. Cost for repairs, lubricating, and cleaning for the working year 1906, 7,440 marks (364*l.*). Sailing radius: unloaded, 13,000 sea miles; loaded, 12,000 sea miles. Mileage in 1905, 21,300 km. (11,472 sea miles); 1906, 22,400 km. (12,073 sea miles).

Russian Oil-Tank Vessel "Sarmat":—

Put in commission in May, 1905, 1,150 tons displacement, 360-430 horsepower. Volga and Caspian Sea. Reversing (Delproposto). Sailing radius at full speed: 14,000 sea miles unloaded; 12,900 sea miles loaded. Mileage in 1906, 19,200 km. (10,348 sea miles).

Russian Oil-Tank Vessel "Djelo":—

5,300 to 5,700 tons displacement, 1,000 to 1,200 horsepower. Made several stormy voyages on the Caspian Sea in the year 1911.

Dutch Oil-Tank Vessel "Vulcanus":—

Put in commission in the summer of 1911. 2,500 tons displacement, 500 horsepower. Fuel consumption for 100 English sea miles, 1,000 kg. to 1,100 kg. (2,204 pounds to 2,425 pounds). Daily crude-oil consumption, 2½ tons for full load. Saving in cost of fuel as against steam engines, 50 per cent. Voyage Rotterdam to Stockholm, in July, 1911: 735 sea miles in 100 hours with cargo of 1,000 tons of oil, and at a speed of 8.5 to 8.9 knots. Fuel consumption, 8½ tons. The *Vulcanus* has already made a very successful voyage of 32 days' duration, to the Black Sea and back, and numerous voyages from Holland to Sweden and England.

(*b*) Tugs.

Russian Tug "Jakut":—

Towing capacity, 4,000 tons; 320 horsepower. The engines have already worked satisfactorily for two years. The maneuvering power is better than with steam engines. The *Jakut* and a steam ice breaker went to the assistance of a ship and towed her out of the ice. On this occasion the fuel consumption of the *Jakut* was 4,380 kg. (9,654 pounds, or 4.3 tons), as compared with 32,500 kg. (71,630 pounds, or 32 tons) by the steamer.

German Tug "Fortschritt," in Hamburg Harbor:—

150 horsepower. Has also made very stormy voyages on the open sea and carried fuel for eight days. Gain in length, one-third over a steamer. Gain in weight of machinery, about one-fourth over steam plant. Weight of fuel only 20 to 25 per cent. of weight of coal for the same power in a steamer.

"Frerichs":—

Towing capacity, 2,200 tons; 200 horsepower. Maneuvering power

same as in a steam vessel. The vessel pays the same insurance premium as a steam vessel. Owing to the absence of boiler and coal bunkers, there is space for the accommodation of twenty passengers. Owing to the absence of the boiler plant the stern of the vessel was made smaller, so that the vessel was more easily handled and the turning circle was diminished. Thirty maneuvers can be carried out in one hour without a change in the air pressure in the starting tanks. With the quantity of air stored in the starting tanks 60 maneuvers can be carried out in succession. A starting tank holds 250 litres (55 gallons), and when completely emptied the pumps can fill it in 15 minutes.

(*c*) Motor Sailing Boats.

"Quevilly":—

About 6,500 tons displacement, 600-700 horsepower on two propellers. The propellers can be uncoupled when using sails only, and their resistance when running light causes a loss of speed of ½ knot. The first ship to cross the Atlantic.

First voyage: Rouen to New York and back, in March, 1911. Engines working during 1,200 hours.

Second voyage: Rouen to New York and back in July and August, 1911. Cost of fuel, 1 dollar per hour per engine. Very satisfactory results.

Third voyage: Havre to New York, lasted 38 days, during 26 days (650 hours) of which the engines worked against a strong wind. The vessel can enter harbors without the help of tugs. The vessel takes up its fuel in New York for the double journey at a price of 32 marks (31s. 4d.) per ton, or 6 pfennig (0.7d.) per e.h.p. hour.

"La France":—

Largest sailing vessel in the world; 10,730 tons displacement, 1,800 to 2,000 horsepower. Sail area, 6,500 sq. m. (69,966 sq. ft.). Will run between France and New Caledonia, and was launched on November 16, 1911.

"Orion":—

Sixty horsepower. Has made voyages since November, 1907, to and from Sweden, Finland, Russia, Germany, and Denmark. The engine is very often used in combination with the sails, and is only in charge of a single carpenter.

(*d*) Merchant Vessels.

(Freight, Passenger, and Combined Freight and Passenger Vessels.)

"Rapp" and "Snapp":—

Two small merchant vessels cruising in Swedish waters. Cargo capacity, 300 tons; 120 horsepower. The engines run for long periods at 55 to 60 revolutions, although the normal number of revolutions is 300. Since 1908 the vessels have made numerous voyages between Sweden, Finland, Germany, Holland, England, Iceland and Norway. On a voyage from the east to the west coast of Sweden, through the canals, seventy-five locks had to be passed, through which the maneuvering power proved to be very satisfactory.

"Cornelis":—

200 horsepower. Has already made numerous voyages between Antwerp and London and Aberdeen. Fuel capacity is sufficient for 32 days.

"Uto":—

Passenger vessel on Lake Zürich. 200 tons displacement, 250-260 horsepower. Has made regular passenger trips on Lake Zürich

since the summer of 1909. Converted steamer. Weight of the previous steam plant, including coal and water, 14,700 kg. (14.46 tons) for 120 km. (64.6 sea miles radius. Weight of the new plant for the double power and 1,200 km. (646 sea miles) radius is 9,750 kg. (9.6 tons). Cost of fuel, one-fourth of the previous cost. Savin labor, one man. Cost of fuel per kilometre, 10.5 cts. (1.6d. per mile).

"Toiler":—

First Diesel sea-going vessel. Cargo capacity, about 3,000 tons; 360 horsepower. The steering is controlled by compressed air. The cabins are warmed by hot water heated by the exhaust from the engines. First voyage from the Tyne to Calais with a cargo of coal was made in the summer of 1911 in very bad weather. Oil consumption, 1.65 to 1.75 tons in twenty-four hours. A steamer of the same size consumes 8 to 9 tons of coal per day. Saving in cost of fuel as compared with steamer, 50 per cent. Gain in cargo capacity, 60 tons. Voyage to North America in September, 1911: Fuel consumption, 2 tons per day. Saving in cost of fuel as compared with steam plant, 11 dollars (45s.); saving in labor, 5 dollars (20s. 10d.) per day. Maneuvering power proved to be very satisfactory.

"Romagna":—

One thousand tons displacement; 800 horsepower. Put in commission September, 1910. Made regular voyages between Ravenna, Trieste and Fiume in the summer of 1911. In consequence of the faulty loading of the cargo the vessel sank in a terrible sirocco in November, 1911.

Vessel of the Hamburg-America Line:—

Nine thousand tons displacement; 3,000 horsepower. Gain in weight and space as compared with a steam plant of the same size, 40 to 50 per cent.

(*e*) Fishing Boats.

"Ewersand":—

German herring boat, built by Frerichs, in Bremen; 254 tons displacement, 90 horsepower. Gain in cargo capacity, as compared with steam fishing boat, 25 per cent. Can take up 750 casks of herrings instead of 600. First voyage in September, 1910, lasted five weeks. Consumption of crude oil, 3 tons, as against 20 tons of coal for steamer. Homeward journey in 75 hours without a stop, in a high sea. On its arrival at the end of October, 1910, 3 tons of crude oil were still unconsumed. Very favorable results, and consequently an order for ten more similar fishing boats was placed.

(*f*) Miscellaneous Boats.

"Fram" (South Polar ship):—

It is the old North Polar ship *Fram* fitted with Diesel engine. Gain through replacing the steam engine by Diesel engine: In engine space, 45 per cent.; weight of engine, 60 per cent.; weight of fuel, 80 per cent.; space for fuel, 85 per cent. Several years' supply of fuel can be stored. Of 380 tons cargo capacity, 100 tons were previously required for the coal storage. The *Fram* sailed for six months from Christiania to the South Polar regions without touching land and without reporting. During the voyage in the Antarctic the engine worked for 2,800 hours without giving any trouble.

(*g*) SUBMARINES.

French Submarine "Aigrette":—

172 tons to 220 tons displacement; 217 horsepower. Long voyage, from Cherbourg to Dunkirk and back, in October, 1908.

"Emeraude":—

390 tons to 450 tons displacement; 600 horsepower. In October, 1908, she made an uninterrupted voyage of 80½ hours on the high seas (1,283 km. or 692 sea miles). The average speed was 9 knots, and the engine worked excellently during the whole voyage. At the beginning of 1909 a long voyage of 800 sea miles was made on the high seas.

"Opale":—

390 tons to 450 tons displacement; 600 horsepower. In September, 1907, made a voyage of 550 sea miles at a speed of 8 m. to 9 m. (26 ft. to 29.5 ft.) per second in difficult navigating conditions. At the beginning of 1909 made a long voyage of 800 sea miles on the high seas. Sailed from Cherbourg to Lorient in 34 hours.

"Circe" and "Calypso":—

350 tons to 490 tons displacement, 300 horsepower. Made a trial trip in the spring of 1910 from Toulon to Nice, Corsica, and back to Toulon, or 600 sea miles in 58 hours. The engines worked excellently.

The experience obtained with Diesel engines on submarines is so satisfactory that today all the navies in the world have, almost without exception, Diesel engines on their submarines. The good results obtained on this kind of vessel were the reason for the fitting of Diesel engines on all other kinds of vessels.

(*h*) WARSHIPS.

Russian Gunboats "Adragan" and "Kars":—

700 tons displacement; 1,000 horsepower. These two vessels made very satisfactory trial trips in the summer of 1911 on the Caspian Sea. Before making these trial trips they sailed from St. Petersburg to the Caspian Sea over the Neva, Lake Ladoga, the canal system and the Volga, arriving on May 19, 1911. It is well known that the Caspian Sea is as stormy as the open sea, and owing to its ice, traveling is considerably more difficult and dangerous for navigation in the winter.

APPENDIX VI.

MOTOR-BOAT SHIPPING IN THE COLONIES.

(Being a summary of the technical portions of a paper read by Dr. R. Diesel, engineer, at a meeting of the Technical Commission of the Committee of Colonial Economy at Berlin on November 13, 1911.)

The technical side of motor-boat shipping is divided into two parts—Boat and Motor.

The African rivers require a special type of flat-bottomed boats, because of their peculiar nature; but there is no difficulty in constructing these boats, and the author has already given many examples of boats on the River Nile and River Congo; on the latter river one boat with 1,300 horsepower, tonnage 450 tons, and only 1.10 m. (3.6 feet) draught. Another boat for the Congo, constructed by the English Bolinder Company, with 40 to 50 horsepower engine, length 14 m. (45.9 feet), width 3 m. to 4 m. (9.8 feet to 13.1 feet), and a speed of 8 knots, can carry a net load of 5 tons. It draws only 0.85 m. to 0.88 m. (2.78 feet to 2.88 feet). By plac-

ing the propellers in a tunnel, sufficient driving power is still obtained, when the propeller is partially immersed, for draughts of 0.5 m. (1.64 feet) and less. Messrs. Cockerill at Seraing constructed a small boat with two protected propellers, for a draught of 35 cm. (1.15 feet). It is provided with winches to float it when aground. A French boat constructor has built a boat with only 10 cm. (4 inches) draught.

The problem of the boats is therefore solved, but not the problem of the engine.

The great importance of navigation for the Colonies is indisputable; the river navigation is actually carried on by steam engines, but they were prevented from great development by the high price of coal. At Stanleyville a ton of marine coal costs 400 francs, which is about twenty to thirty times more than the cost in Europe. This high price makes the working of steam engines with coal prohibitive. A quantity of 30,000 tons of coal per year would be necessary at Leopoldville for the steam vessels at present on the River Congo, but the small-gage railway, Matadi-Leopoldville, can only transport a daily quantity of 50 to 60 tons. It is quite out of the question therefore that the coal stations in the interior can also be supplied with coal. It was proposed to work the vessels with gas engines instead of steam engines; this would mean that only about two-thirds of the coal would be required, but even this small quantity could not be transported by the railway. In addition to that, a special quality of coal would be required, the cost of which would be still considerably higher. The only way to overcome these difficulties was to use wood as fuel. When steam navigation was first started on the Congo it was not difficult to obtain the necessary quantities of wood, but with increased traffic too much wood would be required, and the forests would be destroyed.

If one considers that the 500-ton steamer from Leopoldville to Stanleyville requires 16 tons of wood for each 10-hour voyage, and that it has to take fresh fuel every 10 hours, it is evident that navigation cannot rely on this fuel for the future development and increase. Certainly the price for fuel when using wood was only about 15 centimes (1½d.) per effective horsepower, as compared with 40 centimes (4d.) for gas coal, and 60 centimes (6d.) for steam coal. This price is very low, but, notwithstanding this, wood cannot be the fuel for the future, owing to the foregoing reasons, and steam and gas engines would therefore have to be excluded from working in the colonies.

The question then arises as to the suitability of internal-combustion engines. So far as the author can ascertain there are already some small motor boats with explosion motors used in the colonies, but this cannot be called motor navigation. The danger of the fuel and its high price makes navigation on a large scale impossible for the colonies; it will only be used occasionally on small boats and motor cars.

The only engine to be considered, therefore, is the Diesel, and the author showed in his earlier paper that it is the predestined colonial engine, owing to its development during the last few years and its adoption as a marine engine. It was further shown that it could be worked on any mineral oil. These mineral oils are in all parts of the globe, so that their prices vary very slightly in the different ports, from 40s. to 60s. per ton; for instance, the price for American mineral oils for Diesel engines in London and Great Britain is 43s. per ton, being equal to the price in Belgium and in Scandinavia. Russian oil is delivered to all the Italian ports at a price of 60 francs (48s.) per ton, and this amount is the same in all the other ports of the Mediterranean. In Egypt numerous Diesel engines are working on these oils.

The Diesel engine consumes only about 200 grammes of this oil per effective horsepower hour at an average cost of 0.12d. The consumption of fuel in the Diesel engine is so low that the same number of horsepower can be obtained with a fifteenth part of the weight of the fuel as compared

with wood fuel. If, therefore, Congo steam vessels with 15 tons of wood on board can make a voyage of 10 hours, the same vessel with 15 tons of liquid fuel can make a voyage of 150 hours, besides the saving in time which is effected by not having to call at ports for fresh supplies of fuel.

A vessel fitted with a Diesel engine can take any quantity of liquid fuel on board, which can easily be stored in the bow and stern of the ship where it does not occupy room available for cargo, and it can make long voyages into the interior, taking considerably shorter time than a vessel fitted with steam engines and boilers for wood fuel.

Another very important point is that the costs for working would not be higher in the interior of Africa than near the coast or in Europe, presuming that no duties would be charged for these fuels; it is to be expected that the latter will not be the case, so that this new colonial industry shall have an opportunity of expansion without any restrictions.

To make it clearer the author again states the cost of fuel per effective horsepower hour for the different kinds of fuels mentioned above:

Steam engine with coal fuel, about 60 centimes (6d.).

Gas engine with coal fuel, about 40 centimes (4d.).

Steam engine with wood fuel, about 15 centimes (1½d.).

Diesel engine with mineral oil fuel, about 1 to 2 centimes (1/10d. to 1/5d.).

Another proposal made was to keep the steam engine, but to heat boilers with mineral oil. Some vessels of this kind were built, but the proposition to develop them for the use of the colonies was negatived by the fact that a steam engine of equal horsepower with a Diesel engine consumes about four to five times more mineral oil than the latter.

The great importance of these circumstances for developing the colonies was first recognized by King Albert of Belgium, who had made a journey to the Congo to study the question, and is now working out extensive projects for its colonization. The author had the honor of laying his proposals before the King at an audience, and the result was the order for a large Diesel-engine mail vessel for the Congo River, placed with Messrs. Cockerill at Seraing, which is the first real colonial motor ship. The main dimensions of this vessel are as follows:—Length, 60 m. (196.85 feet); width, 8 m. (26.25 feet); draught, 1.10 m. (3.6 feet), and tonnage, 430 tons. The vessel will have three decks, one of which is intended for the natives, and the other two for the Europeans. The motor power is 1,300 horsepower with two propellers, each exerting 650 horsepower; speed, 25 km. (15.5 miles) per hour. The length of the journey to and from Leopoldville to Stanleyville without taking in fresh fuel is 4,300 km. (2,672 miles). This distance, to and fro, has to be covered in 140 hours, or about fourteen days, whilst the paddle steamers make it in thirty-two days at present. Stock of fuel is 50 tons, and of this stock 15 to 20 tons have to remain at the end of the voyage, so that the vessel is able to supply from its stock smaller boats in the interior which run on tributaries. The cost of this vessel—1,000,000 francs (40,000*l.*)—will be paid by the King himself.

It was stipulated that the natives should be able to take charge of the engines, which necessitated a very simplified trial engine being built first; this engine recently made a successful test. The engines for the Congo mail vessel will now be built after this type.

The example shown by King Albert has had an immediate result by the formation of the Société Anonyme des Petroles du Congo with a capital of 6,000,000 francs (240,000*l.*). The company bound itself to the Belgian Colonial Corporation to lay down petroleum stores along the Congo and its tributaries in which petroleum for at least three months must be kept. The pipe line must have a diameter of at least 10 cm. (4 inches) and the pump engines (eight stations) must be powerful enough to deliver at least 50,000 tons of petroleum to the end of the pipe line.

As a return, the company received from the Congo Government all the land up to 1,000 hectares (2,471 acres) for all the plants, on lease without payment. The concession holds good for fifty years, and after this time all the pipe lines with the material, except the vessels and the stores of petroleum, will pass into the possession of the Congo Government.

The Government reserve the right to buy all the pipe lines after fifteen years, by payment of capital invested in the plant, after depreciation and the reserve funds have been deducted, but with a premium of 33.33 per cent. calculated on the whole amount. The stock of petroleum has to be taken over at cost price, and the petroleum tank vessels at a price fixed by experts.

The selling price of the petroleum is the local price at Matadi Dock, plus 15 per cent. The price at Matadi is fixed between the two parties. The selling price at the station higher up the river than Matadi is fixed in such a way that 15 centimes (1½d.) are added for each kilometer-ton to the Matadi Dock price. These prices hold good for supplies of at least 7 tons per year, and for smaller supplies an increase of 10 per cent. can be obtained. If the net profit does not come up to 7½ per cent. within two years, after having made all deductions and depreciation, the price can be increased 1 centime per kilometre-ton, and it must be decreased the same amount if the net profit is more than 7½ per cent. The petrol supply for the Congo is thus secured, and the first condition—the supply of the necessary fuel—is fulfilled.

The calculations for the pipe line have been based on the fact that the consumption of wood fuel higher up the Congo is equivalent to 20,000 to 22,000 tons of petroleum yearly; but it is certain that at the beginning this large quantity of petroleum will not be wanted, especially during the experimental period, and because motor navigation on the Congo itself can only be taken into consideration, as the navigation on the tributaries will still use wood fuel.

The vessels at present in use, about 4,000 horsepower, will also work on wood fuel until the Diesel-engine vessels are supplied. The locomotives of the Matadi and Leopoldville Railway will also work on petroleum in the future, and experiments in this direction have already been made on three locomotives. It is also anticipated that the existing petroleum piping will be the reason for the erection of a good number of stationary and agricultural Diesel engines, and thus there will be new customers for petroleum.

Considering the conditions given above, the price for the mineral oil per ton would be—

	fr.		s.	d.
At Matadi Quay, about	75	=	60	0
Plus 15 per cent., about	11	=	8	10
	86	=	68	10
400 km. higher up in the country an addition of 400 × 15–60 fr.		=	48	0
Therefore price per ton	146	=	116	10

Numerous Diesel engines are working in many European countries where the prices for mineral oils are the same owing to the high duty which has to be paid. But navigation on the main streams would be independent of this increase in price, as the vessels could take any amount of fuel on board at the beginning of their voyage, sufficient for the journey to and fro.

As already stated, the large vessels on the main rivers can give to the ships in the interior their surplus fuel; thus cheap navigation can be carried on along the rivers furthest in the interior. From this it follows

that, where there are rivers in the colonies, motor navigation ought to be started first, and thus the very expensive construction of railway lines will be obviated. Important districts without rivers would get small railways to connect them, and the locomotives can also be driven by Diesel engines.

The construction of the colonial Diesel engine ought to be as simple as possible, and only very few patterns of cylinders and other parts ought to be built, so that it would be possible to exchange and replace single parts easily; larger units ought to be constructed by adding two, three, four and more cylinders. The engine working with many cylinders is not a drawback, but a good feature, because the engine can continue working even if one cylinder stops; another feature connected with the simplicity of construction is that only very few spare parts are required, and that the same spare parts can be used for all the cylinders. These spare parts can be stocked all over the colony, and would prove to be very useful if all the Diesel engines in one district were of the same pattern. This, however, can only be considered when the Diesel engine has become well introduced. The importance of the possibility of using vegetable oils in the Diesel engine has already been discussed in the paper.—"Engineering."

THE WIDER ADOPTION AND STANDARDIZATION OF WATER-TUBE BOILERS.*

By E. M. Speakman.

Ten or fifteen years ago no broad technical questions absorbed the attention of marine engineers to the extent which that of the water-tube boiler did, and probably few have been more keenly discussed. The subject grew in interest, and criticism of their adoption continued to become more intense from the time of the decision to install Belleville boilers in H.M. cruisers *Powerful* and *Terrible* till the appointment by the Admiralty, in 1900, of a committee of independent engineers to investigate and report on the status and future of naval boilers. Partly, perhaps, owing to the final verdict of this committee being rather of the nature of a recommendation to wait and see what the results of experience with newer and modified designs should be, and partly to fresh interests arising in other sections of propelling machinery, public interest in the water-tube boiler has recently somewhat flagged. This is probably, to a large extent, attributable to the greatly increased reliability and freedom from the troubles and breakdowns which characterized the boilers of earlier years, and which thereby attracted much attention from the technical press. Except for cases of culpable misuse, nearly all the difficulties that originally arose with water-tube boilers were due—

1. To bad design arising from inexperience;
2. To inferior workmanship, and the employment of unsuitable tubing; and
3. From a desire to press the advantages of reduced weight and capability to withstand forcing to an undue extent.

Compared with modern designs, the water-tube boiler of twelve years ago was in a strikingly similar position to the motor car of the same period, or to the flying machine of today. But notwithstanding its defects, its efficiency in those days was distinctly good, and the improvements that have taken place in the last few years have been mostly in the direction of reliability and durability—that is, in better mechanical design, coupled

* Paper read before the Institution of Engineers and Shipbuilders in Scotland on February 20.

with use of better material and improved proportions, which, although adding somewhat to the weight, have not appreciably increased the thermal efficiency in comparison to the extent to which the troubles have been removed.

Concurrently with, and as a result of, improvement in mechanical design, there has been a constant tendency for the various types of the last decade to become standardized into more reasonable shapes and proportions, the Thornycroft, Normand, Reed and other designs being now largely merged into the prevailing Yarrow type, which alone has retained its original characteristics. Of the various large-tube boilers, the Babcock & Wilcox, Belleville and Niclausse still remain, but in improved form. Practically all the above, with the exception of the Babcock & Wilcox, are, at the present time, naval boilers only, and their general adoption, both in the large and small vessels of the various navies of today, is shown in Table I.

TABLE I.

Nation.	Type of boiler adopted.		
	Large war vessels.	Small war vessels.	Torpedo craft.
England	Yarrow.	Yarrow.	Yarrow.
	Babcock & Wilcox.	.....	White-Forster.
France*	Belleville.	Lagrafel-D'Allest.	Normand.
	Niclausse.	Du Temple.	White-Forster.
Germany.....	Schulz-Thornycroft.	Schulz-Thornycroft.	Schulz-Thornycroft.
Italy*†	Babcock & Wilcox.	Cylindrical.	Thornycroft.
	Belleville.	.....	.....
	Niclausse.	.....	.....
Japan	Belleville.	Miyabara, etc.	Yarrow.
	Miyabara.	.....	.....
United States*	Babcock & Wilcox.	Babcock & Wilcox.	Normand and other express types.

* Very few small cruisers have been built by these Powers in the last ten years.
† Italy is adopting Yarrow boilers in the latest battleships, as are also Russia and Spain.

Throughout the merchant service the cylindrical boiler is, of course, still generally adopted, and for all ships in which the weight of boilers is a relatively small fraction of the total displacement no other type seems likely to displace it. There are, however, distinct signs that, for certain special services, the wider adoption of water-tube boilers would be a welcome innovation.

Referring to Table I, it is significant that only one Power has so far adopted a standard type of boiler; the German Admiralty apparently rightly regards a boiler as a steam producer of a certain efficiency in relation to its weight, and as such it is independent of size or type of ship.

Although in reintroducing the subject of water-tube boilers from the point of view of wider adoption and standardization, it may be felt that so much has already been said about boilers that there is little new to add, there are, however, various matters pertaining to them that may afford ample basis for discussion. Although to some extent these may relate to the design *per se*, at the same time others tend to affect ships in their entirety, not only as regards the present day, but probably much more so in the near future. The reasons for and against the wider adoption of water-tube boilers in vessels to which, while appearing applicable at first sight, they have not yet been applied, are much the same as those quoted fifteen years ago in making comparisons with cylindrical boilers; but it should be remembered that the conditions now met with are very different.

As a steam producer the water-tube boiler is now of proved reliability, durability and efficiency, instead of becoming a doubtful quantity; and its light weight, and great capacity for steam production in relation thereto, strongly recommend it in many cases beside those of naval work. Compared with their dimensions fifteen years ago, the size of ships on many services is now approaching a commercial maximum—even if that be only temporary—either in length or draught of water; and, again, on limited dimensions—*i. e.*, weight—there is a constantly increased demand for speed. For the bulk of mercantile tonnage, much of which is in relatively slow vessels, the water-tube boiler, unless more efficient than the cylindrical type, and no more costly either originally or in maintenance, cannot be considered commercially; and even if it complied with these conditions, other practical circumstances arise, peculiar only to the nature of the service on which such vessels are employed, that would tend to limit, if not to exclude, its adoption. For all vessels comprised in the cargo-carrying and low-speed types, the weight of machinery is very small in relation to the displacement. It is only when this ratio becomes high that the introduction of water-tube boilers should be carefully studied; and it it just in these cases that limitations affecting the ship as a whole—length or draught of water, for instance—are seriously beginning to be felt. Their scope for wider adoption, consequently, is still somewhat circumscribed.

In Table II is given a list showing the percentage of boiler weights to normal displacement for a number of typical vessels. The weight referred to, in order to exclude all variables except the generator portion only, consists of the boiler complete with water to normal working level, lagging and fire-bars, but no uptakes or mountings. It will be seen at a glance that it is only in the cases in which the percentage forms a large proportion of the displacement that the saving in weight due to installing water-tube boilers would be justified. A few examples of naval vessels, all of which are fitted with Yarrow boilers, are added for comparison. From the same table will be seen the types of vessels in which the percentage weight is exceptional, the maximum condition being, of course, obtained in vessels of the destroyer type, when the horsepower per ton of displacement may be as much as 27. In small cruisers it may be as much as 8, but the figure rarely exceeds 4, even for merchant vessels of high speed and small size. In the fastest Atlantic liners this falls to about 2; in slow carrying vessels it may be only one-tenth of this. With the exception of river steamers, in which shallow draught is essential, these figures indicate that only in very high-speed Atlantic liners and fast Channel steamers would the light weight of water-tube boilers tend to counterbalance certain practical objections to them. In the former case the enormous growth of length and breadth has only been accompanied by a relatively small increase in draught of water, and at the present day there are a large number of vessels on the Europe to New York service which are unable to take full advantage of the draught to which they might safely be loaded, on account of the depth of water at their ports of departure. For the time being the practical limit is about 34 feet. By the substitution of Yarrow boilers for Scotch, a fast ship, 800 feet by 90 feet, might carry nearly 1,500 tons more cargo, or have her draught reduced by 12 inches. In Channel steamers the same considerations apply; length is limited on the French routes, draught on the Ostend route, and high speeds are essential on all. In fact, on the limited maximum dimensions permissible in each case the attainment of a higher power on less weight, and in less space, is the only means of obtaining a much faster ship.

Now, although the water-tube boiler might appear to offer the solution, there are various practical difficulties still confronting its adoption. The greater initial cost might alone be considered sufficient objection if it were not for the advantages its light weight confers. But its liability to prime in the event of condensers leaking, the tendency of the tubes to

TABLE II.

Type of ship.	Speed, knots.	Displacement, tons.	Number of boilers.	Type (cylindrical).	Pressure, pounds.	Weight of boilers / Displacement × 100
Channel steamer :						
Turbine	23	2,700	8	S.-E.	185	19.45
Reciprocating	22	2,200	4	D.-E.	175	17.7
Turbine*	20	2,000	2	D.-E.	160	15.4
Paddle	20	1,500	6	S.-E.	120	18.0
Reciprocating	16	1,600	2	D.-E.	180	10.9
River steamer :						
Turbine	21	750	1	D.-E.	150	18.65
Reciprocating	12	1,000	2	D.-E.	160	9.0
Merchant steamer	10	9,000	2	S.-E.	180	2.15
Do. †	12	9,000	4	S.-E.	180	3.31 †
Do. †	12	9,000	4	S.-E.	215	3.62 †
Do.	13	12,000	3	S.-E.	180	3.45
Do.	14	10,500	6	S.-E.	190	4.67
Do.	15	10,700	6	S.-E.	180	4.8
Combined cargo and passenger.	17 (at sea)	20,000	6	D.-E.	215	7.2
	18 (trial)	17,000	6	S.-E.		8.5
Do.	17.5 (trial)	15,200	4	D.-E.	215	5.8
			2	S.-E.		
Passenger steamer :						
Turbine	20	10,500	6	D.-E.	175	11.0
Atlantic liner : Turbine	25.5	36,000	23	D.-E.	215	11.0
			2	S.-E.		
	23	60,000	18	D.-E.	215	5.4
Battleship	22	22,000	18	Yarrow.	220	2.1
Cruiser	24	3,000	10	Yarrow.	220	6.5
Destroyer	36	2,000	10	Yarrow.	220	18.0
Do.	33	900	5	Yarrow.	220	21.0
Do.	30	450	4	Yarrow.	220	16.0

* In a similar vessel to this, now under construction, but fitted with five Babcock boilers, the proportion of weight of boilers has been reduced to 8.5 per cent.

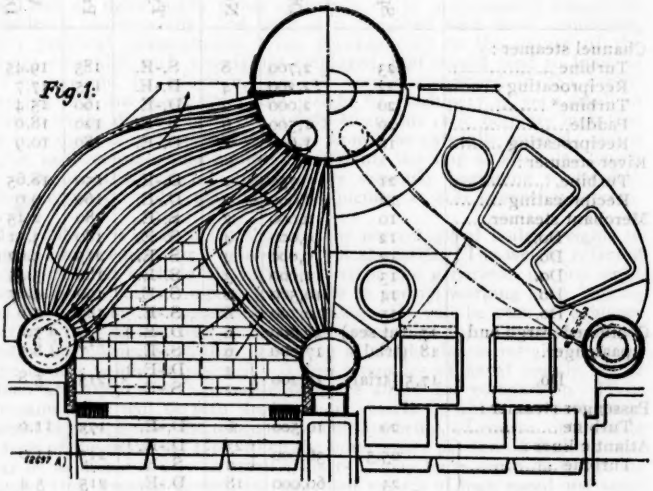
† Note influence on percentage weight due to higher pressure used for quadruple in place of triple-expansion machinery.

become choked externally more rapidly than those of the cylindrical type, and the difficulty of the necessarily frequent tube cleaning, together with the distinctly and rather unnecessarily drastic survey requirements demanded, are also features, as far as continuous merchant-service work is concerned, that do not commend it to shipowners and their marine superintendents. Further, there is the possible slightly lower efficiency, compared with the large cylindrical boiler.

Against all this there is the fact that, for the same hourly steam production, which, other things being equal, is the true criterion of merit, the three-drum type of water-tube boiler weighs less than half what the cylindrical boiler does. When this proportion of saved weight forms 5 to 10 per cent. of the complete displacement of the ship, it becomes an important question as to whether the alleged disadvantages are not more

than counterbalanced in view of the saving in the ship as a whole. It is not suggested that an average naval boiler should be installed without change into merchant vessels, but there are features connected with the design thereof that strengthen the idea that water-tube boilers, as used in warships, may be installed in fast mercantile ships if, as seems easily possible, they can be so modified in minor ways as to render them thoroughly serviceable for the practical and economical requirements involved, whilst simultaneously considerably reducing the above-mentioned objections. These points may be considered in detail.

Fig. 1.



Type of Boiler.—The Boiler Committee appointed by the Admiralty in 1900 investigated no less than thirty-six different types of water-tube boilers, only three of which now find a place in modern British practice. These types are:

I. Yarrow, applied to large and small vessels.
II. Babcock & Wilcox, applied to large vessels only.
III. White-Forster, applied to small vessels only.

With the possible exception of the German Navy, where the Schulz boiler, as illustrated* in Fig. 1, is now universally adopted in battleships and destroyers, the British Navy has attained a degree of standardization in boiler practice not found elsewhere. The United States Navy Department, while adhering to Babcock & Wilcox boilers for big ships, adopts several types for destroyers, as does also the French Admiralty, which retains both Belleville and Niclausse boilers for battleships. This divergence in practice is not easy to understand. The inconsistency displayed in the case of the four principal navies enforces the idea that the separate authorities are willing to accept *pros* and *cons* entirely at variance with those of the others, the reliance on individual views and corresponding policies being based presumably on the developed service opinion of many years' growth. For wider adoption this hardly facilitates the choice of prospective users, and it is, in the author's view, somewhat surprising

* The German Admiralty oil-fuel boiler is of the ordinary three-drum type.

that, after so many years of use, a closer consensus of opinion has not been more generally achieved.

Obvious disadvantages would seem to be involved in those types wherein excessively curved tubes, or forms of tube-baffling, are adopted, on account of difficulties of manufacture, of cleaning internally and externally, and also of rapid repair. There is the further great objection that any considerable curving towards the fire bars (Figs. 1 and 3) tends to reduce the combustion space. On this basis straight, or nearly straight, tubes would seem to be necessary, and in spite of apparent success in other types, practical experience of both strongly emphasizes the objection to sharp bends.

Fig. 2.

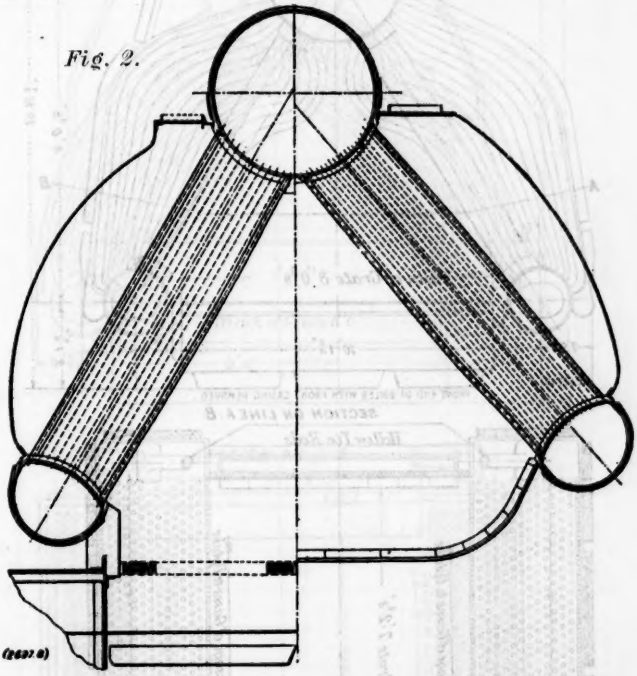


Fig. 2 shows two half-sections of typical Yarrow boilers, one of the 1¼-inch tube type, as used on large vessels with coal fuel, the other with 1⅛-inch tubes, designed for oil burning. The simplicity of this design for manufacture and cleaning compared with Figs. 1, 3 and 4 is undeniable. In Fig. 3 is shown the Normand boiler, as used in the United States cruiser *Chester*. Fig. 4 shows a variation of the Du Temple type, as fitted to her sister ships *Salem* and *Birmingham*. The extreme curvature of the tubes and the tube baffling adopted not only lessen the combustion space for given over-all dimensions, but also prevent the best use being made of the heating surface installed.

Capacity for Steam Production.—Generally speaking, a large boiler is proportionally more economical than a small one, and a double-ended

Fig. 3.

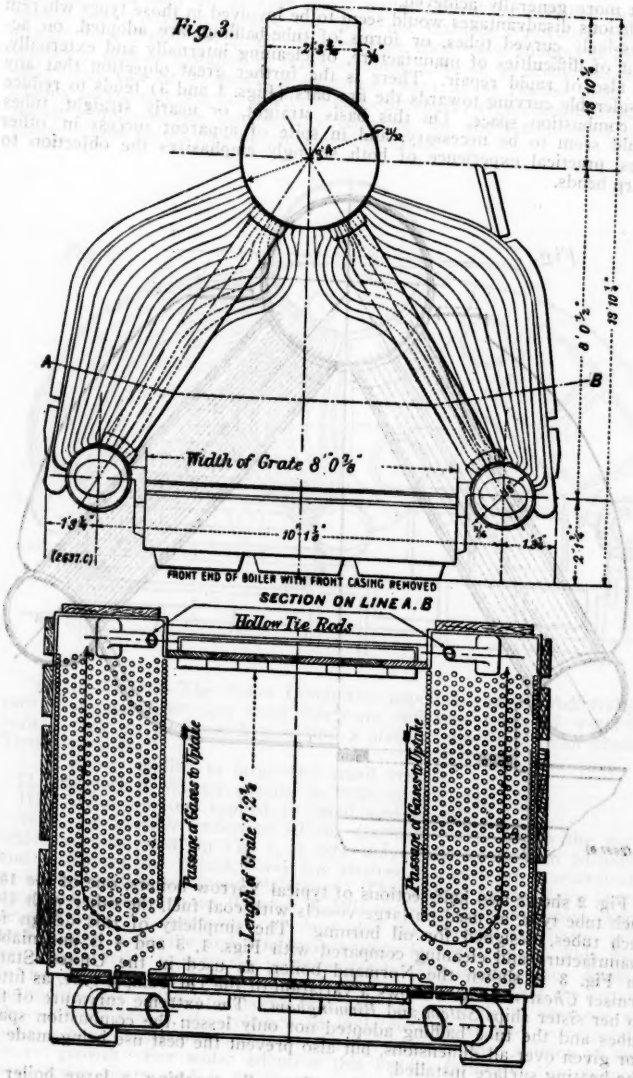
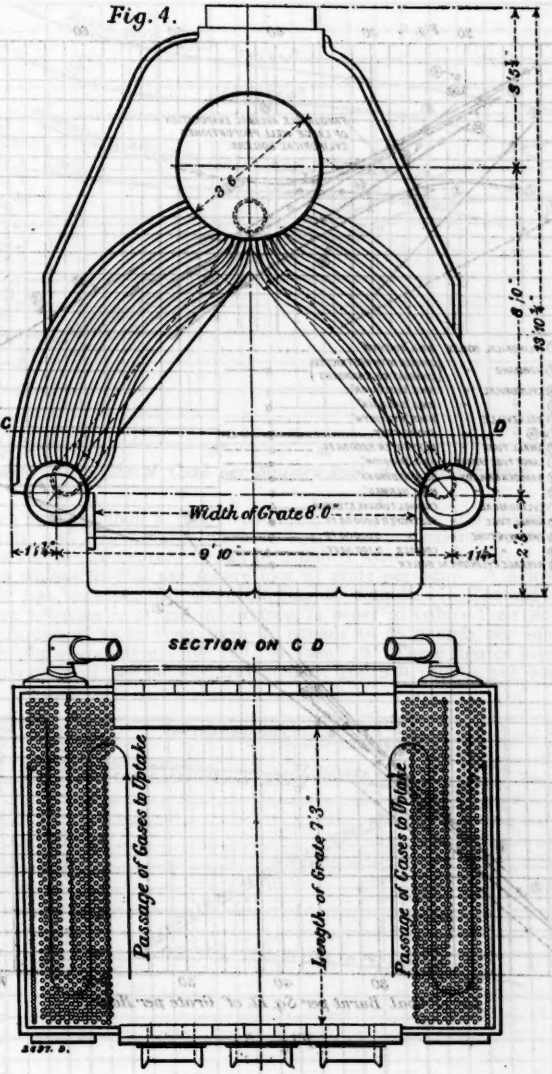

Width of Grate 8'0⅜"
FRONT END OF BOILER WITH FRONT CASING REMOVED
SECTION ON LINE A. B
Hollow Tie Rods
Passage of Gases to Plate
Length of Grate 7'2⅝"
Passage of Gases to Plate

Fig. 4.


3'6"
C
D
Width of Grate 8'0"
1'1½"
9'10"
1½"
2'6"
6'10"
SECTION ON C D
Passage of Gases to Uptake
Length of Grate 7'3"
Passage of Gases to Uptake

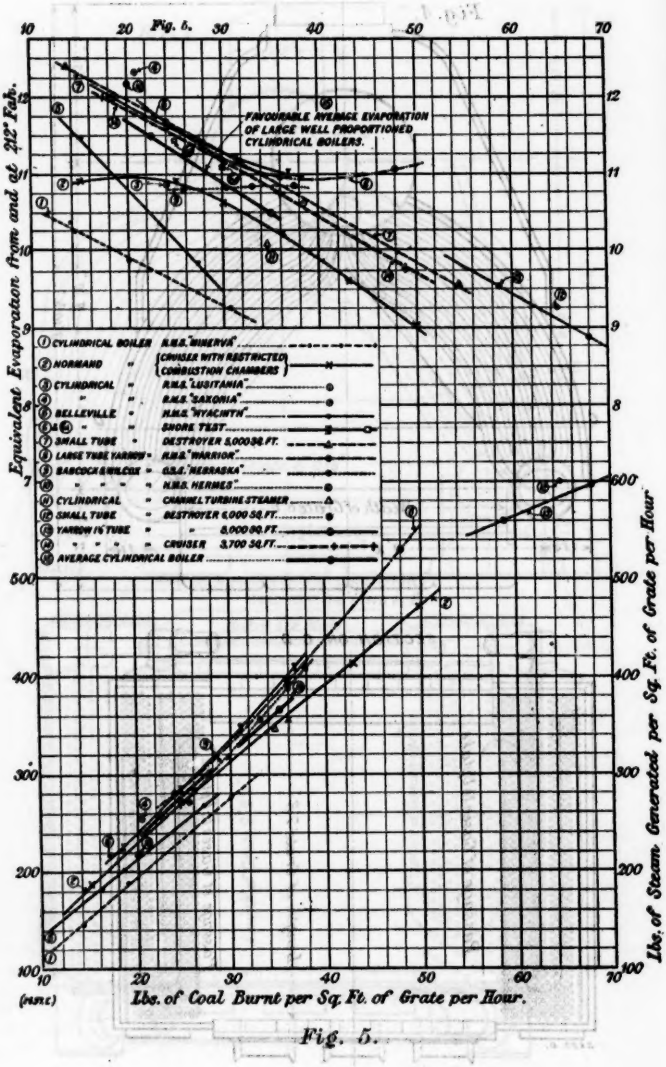
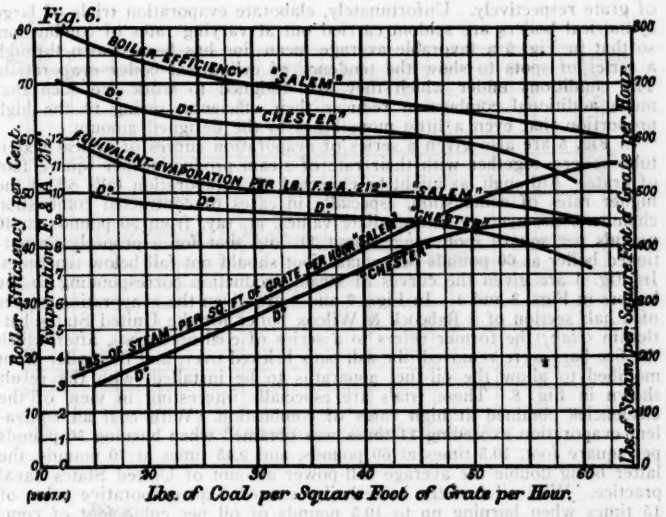


Fig. 5.
FAVOURABLE AVERAGE EVAPORATION OF LARGE WELL PROPORTIONED CYLINDRICAL BOILERS.
Equivalent Evaporation from and at 212° Fah.
Lbs. of Steam Generated per Sq. Ft. of Grate per Hour.
Lbs. of Coal Burnt per Sq. Ft. of Grate per Hour.
(nmc)
① CYLINDRICAL BOILER H.M.S. "MINERVA"
② NORMAND " (CRUISER WITH RESTRICTED COMBUSTION CHAMBERS)
③ CYLINDRICAL " R.M.S. "LUSITANIA"
④ " " R.M.S. "SAXONIA"
⑤ BELLEVILLE " H.M.S. "HYACINTH"
⑥ ⑧ " " SHORE TEST
⑦ SMALL TUBE " DESTROYER 5,000 SQ. FT.
⑧ LARGE TUBE YARROW " H.M.S. "WARRIOR"
⑨ BABCOCK & WILCOX " U.S.S. "NEBRASKA"
⑩ " " " H.M.S. HERMES"
⑪ CYLINDRICAL " CHANNEL TURBINE STEAMER
⑫ SMALL TUBE " DESTROYER 6,000 SQ. FT.
⑬ YARROW 16 TUBE " " 8,000 SQ. FT.
⑭ " " " CRUISER 3,700 SQ. FT.
⑮ AVERAGE CYLINDRICAL BOILER

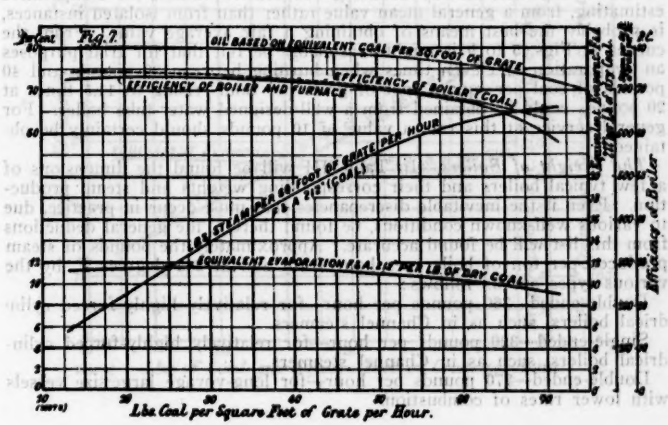
Fig. 5.

Fig. 6.



boiler more so than a single-ended one. Experience has shown that for Atlantic and long-distance work an equivalent evaporation of eleven times from and at 212 degrees F. can be attained when burning from 23 pounds to 26 pounds of coal per square foot of grate per hour. A figure as high as 12 has been reached when burning 20 pounds. In Channel steamers, where economy is of less importance, in order to reduce their size, boilers are forced so as to burn from 32 pounds to 38 pounds per square foot, when the evaporative value falls to about 10. These figures represent an evaporation of 270 pounds, 240 pounds, and 350 pounds per square foot

Fig. 7.


OIL BASED ON EQUIVALENT COAL PER SQ. FOOT OF GRATE
EFFICIENCY OF BOILER AND FURNACE
EFFICIENCY OF BOILER (COAL)
LBS. STEAM PER SQ. FOOT OF GRATE PER HOUR F & A 212° (COAL)
EQUIVALENT EVAPORATION F & A 212° PER LB OF DRY COAL
Efficiency of Boiler
Lbs. Coal per Square Foot of Grate per Hour.

of grate respectively. Unfortunately, elaborate evaporation trials of large cylindrical boilers are seldom carried out at varying rates of combustion, so that in Fig. 5 a favorable average mean line has been drawn through a series of spots to show the tendency of cylindrical boiler evaporation. The conditions under which they are designed to work are such that much additional combustion reduces their efficiency, owing to the high proportion that even a little more bears to the designed amount.

In Fig. 5 are also given a series of evaporation curves of typical water-tube boilers, together with their rate of steam production per square foot of grate. Although, as might be expected, the evaporation falls off at the higher rates of combustion, especially in cases of contracted combustion chambers, the average intermediate values, at, say, from 30 pounds to 40 pounds per square foot, range about 11, and that for a properly proportioned boiler at 60 pounds per square foot should not fall below ten times. In Fig. 6 are given the curves of steam production corresponding to the boilers in Figs. 3 and 4. In Figs. 7 and 8 are given the evaporation curves of a half section of a Babcock & Wilcox boiler for the United States battleship *Utah;* the former refers to a series of coal-fired tests, after which the fire bars were removed, the ash pans bricked over, and the boiler front modified to allow the oil-fuel apparatus to be installed, with the result shown in Fig. 8. These trials are especially interesting in view of the efficiencies obtained at high rates of combustion. With coal an equivalent evaporation exceeding 11 times was obtained when burning 50 pounds per square foot, 10.5 times at 60 pounds, and 9.45 times at 70 pounds, the latter being double the average full-power amount of United States naval practice. With oil fuel the same boiler exceeded an evaporative value of 15 times when burning up to 10.5 pounds of oil per cubic foot of combustion space; when burning 15 pounds the evaporation was 13 times, which is equal to 16.45 pounds of steam per square foot of heating surface per hour. The relative efficiencies, Fig. 7, when burning coal at 70 pounds per square foot, and the equivalent amount in oil, are 60 and 72.6 per cent. respectively. Fig. 9 gives the evaporation curves of a Mosher type of boiler built to replace the cylindrical boilers of the United States battleship *Kearsarge.* No little difficulty and cost are involved in conducting large series of evaporative trials, and local conditions frequently affect the readings to a considerable extent. By massing a varying series of curves together, and by deducing evaporative values, for comparative estimating, from a general mean value rather than from isolated instances, is probably the best means of obtaining a fair average value. From the curves in Figs. 5 to 9 it might reasonably be felt that for trial purposes an evaporation of eleven times when burning between 30 pounds and 40 pounds of coal per square foot of grate per hour rising to 11.5 times at 20 pounds could be obtained from a well-designed water-tube boiler. For general service at this rate a value of 10 pounds should certainly be obtained.

The Weight of Boilers.—In Table III will be found the dimensions of a few typical boilers and their corresponding weights and steam production. Even if the inevitable discrepancies that must occur in practice, due to various well-known conditions, be found therein, the general deductions from this list will be found accurate. Approximately the pounds of steam produced per ton of boiler per hour from and at 212 degrees F. by the various types are as follows:

Double-ended—360 pounds per hour—for relatively highly-forced cylindrical boilers, such as in Channel steamers.

Single-ended—320 pounds per hour—for relatively highly-forced cylindrical boilers, such as in Channel steamers.

Double-ended—270 pounds per hour—for long-voyage large-size vessels with lower rates of combustion.

Fig. 8.

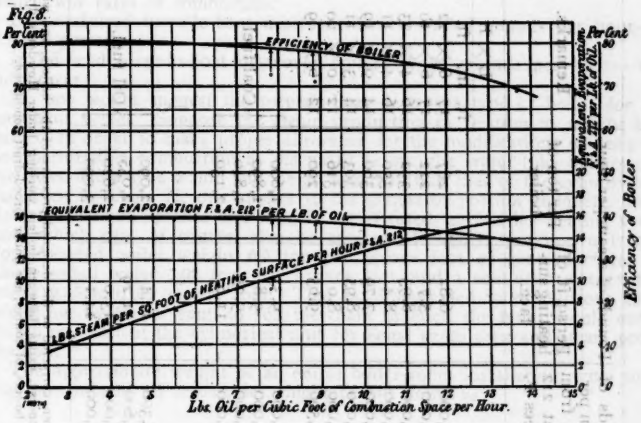

Per Cent
EFFICIENCY OF BOILER
EQUIVALENT EVAPORATION F. & A. 212° PER LB. OF OIL
LBS. STEAM PER SQ. FOOT OF HEATING SURFACE PER HOUR F. & A. 212°
Equivalent Evaporation F. & A. 212° per Lb. of Oil.
Per Cent
Efficiency of Boiler
Lbs. Oil per Cubic Foot of Combustion Space per Hour.

Fig. 9.

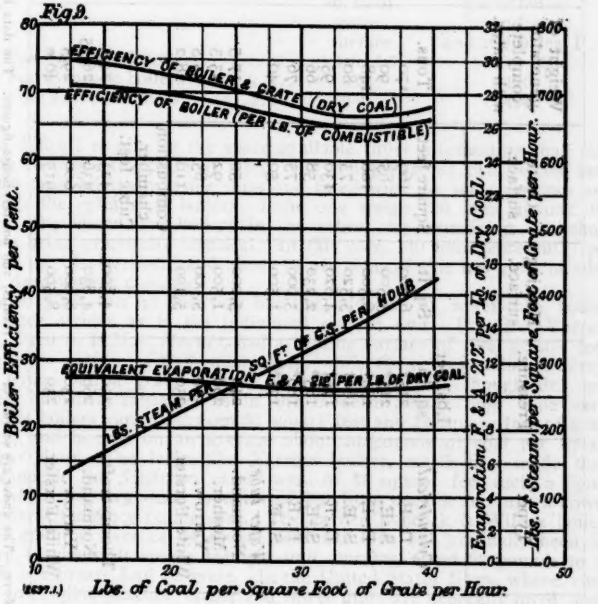

EFFICIENCY OF BOILER & GRATE (DRY COAL)
EFFICIENCY OF BOILER (PER LB. OF COMBUSTIBLE)
EQUIVALENT EVAPORATION F. & A. 212° PER LB. OF DRY COAL
LBS. STEAM PER SQ. FT. OF G.S. PER HOUR
Boiler Efficiency per Cent.
Evaporation F. & A. 212° per Lb. of Dry Coal.
Lbs. of Steam per Square Foot of Grate per Hour.
Lbs. of Coal per Square Foot of Grate per Hour.

TABLE III.

No.	Type.	Pressure.	Heating surface.	Grate surface.	Weight of generator complete with water.	Pounds of steam per hour, from and at 212 degrees F.	Pounds of steam per hour.		Remarks.
							Per sq. ft. of heating surface.	Per ton of boiler.	
		Lbs. per sq. in.	Sq. ft.	Square feet.	Tons.				L × B ft. in. × ft. in.
	Cylindrical.								
1	D.-E.	215*	6,600	169	170	45,000	6.82	267	17 6 × 22 0
2	S.-E.	215*	3,500	85	90	20,000	5.71	222	17 6 × 10 7½
3	D.-E.	160†	6,500	168	148	58,000	8.93	392	16 4 × 23 0
4	S.-E.	160†	3,320	73	80	28,000	8.4	350	16 4 × 11 6
5	D.-E.	175†	4,620	130	95	36,000	7.79	380	14 0 × 19 0
6	S.-E.	185*	2,230	58	66	20,000	8.95	303	14 3 × 11 6
7	D.-E.	180*	3,000	75	76	24,000	8.0	316	12 0 × 20 0
8	S.-E.	175*	1,500	50	40	14,500	9.65	290	12 6 × 9 3
	Water tube.								
9	Normand.	240	3,000	50	17.5	28,000	9.33	1,600	Coal fuel.
10	Mosher.	260	1,500	92	23.3	44,000	9.8	1,890	
11	Yarrow.	230	5,000	85	25.5	47,000	9.25	1,840	
12	White-Forster.	230	5,100	110	28.2	60,000	11.6	2,130	
				Combustion chamber. Cubic feet.					
13	Thornycroft.	250	4,500	423	24.5	51,300	11.4	2,090	Oil fuel.
14	Normand.	260	4,830	336	26.75	54,600	11.29	2,035	
15	Yarrow.	230	6,000	600	29.5	78,000	13.0	2,620	
16	White-Forster.	230	8,500	915	40.0	120,000	14.1	3,000	

NOTE.—The above are service or trial results, and not test-house figures. The data have been corrected to the designed pressure of the boilers, and where necessary, for local conditions of working. A considerable difference in evaporative value, of course, exists between boilers working under Howden's draft,* closed-stokehold† conditions, or under natural draft; all the data given of torpedo-boat destroyer boilers, of course, are under forced-draft conditions.

Single-ended—230 pounds per hour—for long-voyage large-size vessels with lower rates of combustion.

For coal-fired torpedo-boat destroyer boilers—1,800 pounds per hour—At full power.

In oil-fired torpedo-boat destroyer boilers—2,400 pounds per hour—At full power.

No one would suggest the torpedo-boat destroyer type of boiler for a merchant vessel. Suppose the steam production in relation to weight be halved, in order to make proper allowance for the modifications to design, and permissible combustion as discussed below, the water-tube boilers of the three-drum type would still weigh less than half that of the cylindrical. As given in Table II, this applies to the generator portion complete, and does not include uptakes and funnels, or, if fitted, a Howden heating system, which can, of course, also be applied. Roughly speaking, a large double-ended boiler weighs 0.9 ton per square foot of grate surface, a single-ended boiler 1.02 tons per square foot, and a coal-fired water-tube boiler 0.35 ton per square foot. That is, if increased in proportions and scantlings by 50 per cent. to suit merchant practice, the latter would only weigh one-half of the cylindrical unit for equal grate area and steam production.

The approximate weight of an entire boiler-room installation, with piping, pumps, and all accessories, might be put down as:

	Tons per sq. ft. of grate surface.	Pounds of steam per hour per ton of boiler room from and at 212 deg. F.
For cylindrical boilers.....	1.4 to 1.7	150 to 200
For Babcock boilers.....	0.65	450
For Yarrow boilers.....	0.9	490

It is difficult to give in the space available fuller information than that tending to prove the point at issue; but in the case of two vessels each of 2,000 tons—virtually sister ships and fitted with the same turbines and shafting—the cylindrical boilers of the one weigh 320 tons, against 180 tons of the water-tube boilers in the other, the remaining stokehold weights being practically identical. In this case, 140 tons represent 7 per cent. of the displacement, and about 70 per cent. of this has gone in additional dead-weight capacity.

Size of Unit.—In no direction has the advance of water-tube boilers made such strides as in the individual size of unit. The first Yarrow boiler, fitted in H.M.S. *Hornet,* had a heating surface of 1,027 square feet, and a grate surface of 20.5 square feet; while the corresponding Thornycroft boiler had a grate surface of 47.3 square feet, though this was divided into two furnaces, as in the modern Schulz type. The early Belleville boilers varied between 53 square feet and 58 square feet of grate surface, and burned on an average about 26 pounds of coal per square foot per hour. The large-tube Yarrow boilers, which first made their appearance in the *Swiftsure* class,* were of 53 square feet each, a figure from which the large-tube type varied but little until it recently followed the general tendency to increase. The earlier Babcock & Wilcox boilers in this country were of 64 square feet, and this has gradually been increased to about 90 square feet per unit, burning about 22 pounds to 24 pounds per square foot of grate. In the United States Navy, where, compared to English practice, larger and fewer units are generally fitted, some of the earlier Babcock & Wilcox boilers were of 96 square feet, rising

* H.M.S. *Medea* (see Boiler Committee's Report) excepted.

to 103 square feet in the *Delaware* class, and to 119 square feet in the *Utah* class, and burning from 30 pounds to 36 pounds of coal per square foot of grate per hour. In the small-tube three-drum type of twelve years ago from 55 to 60 square feet was common. More recent boilers for small craft have risen first to 70 square feet and later to 85 square feet. In the recent United States torpedo-boat destroyers 90 square feet is common; in the German torpedo-boat destroyers 100 square feet, and, in one case, 125 square feet in a two-furnace unit has been adopted. This boiler is about 22 feet wide over the lower drums. The largest individual coal grate of the author's experience in this country is in a White-Forster boiler recently built at Birkenhead for a foreign torpedo-boat destroyer, and which has 111 square feet. Messrs. Yarrow's double-ended boilers for the Brazilian torpedo-boat destroyers of the *Matto Grosso* class exceeded this, but in their case the furnace was completely divided transversely into two separate parts by a brick wall, and the individual grates of this double-ended boiler did not exceed 70 square feet. These exceptional areas, except for the double-ended type, are quite outside mercantile use, but are quoted as examples of limiting size in coal units. A reasonable average of from 70 square feet to 80 square feet of grate in a three-drum type is a feasible project in small mercantile vessels, but for economy the coal burnt per square foot of grate per hour should be between the economical limits of the large and small-tube types, say from 30 pounds to 40 pounds per square foot, although these boilers at full power burn easily 25 per cent. more for long periods. Strictly speaking, the size of coal-fired units rests entirely on the maintenance of clean, level fires on a large grate; 120 square feet is exceptionally large, and is only possible with most systematic stoking and picked fuel. Even so, no difficulty has been experienced in keeping good fires either on 110 square feet on one grate or 125 square feet on a Schulz grate when burning about 65 pounds per square foot of grate per hour. Such a consumption is only possible for a short period, say, from 6 to 8 hours, without considerable cleaning, and for runs of 100 hours (an Atlantic passage) a much lower consumption per square foot would, of course, require to be adopted. Obviously, for short-run routes larger grates might be adopted than on the ocean services.

When oil fuel is used instead of coal the question of firing is of vastly less importance and greatly lessens the limitation of size of unit. In fact, this might almost then be determined by the loss of power due to an accident to one unit forming too great a proportion of the evaporating plant. Oil-fired boilers of from 6,000 square feet to 7,000 square feet of heating surface are now common. The largest individual boiler of which the author has experience is the unit in the Argentine torpedo-boat destroyers of the *San Luis* class, built by Messrs. Cammell Laird & Co. at Birkenhead, which has a heating surface of 8,500 square feet. It is of the single-ended White-Forster type, and has a combustion space, of 915 cubic feet. With 5 inches air pressure it has generated from feed water at 212 degrees F., about 120,000 pounds of steam per hour at 230 pounds pressure, without the slightest difficulty of operation or adverse effect on the boiler. The two tube rows nearest the fire are 1⅛ inches in diameter, the twenty others are 1 inch in diameter; the upper drum is 4 feet 8 inches in diameter. Transversely over all the boiler measures 19 feet. Probably this is the largest boiler yet made, but experience of it indicates that oil-burning units having from 10,000 square feet to 12,000 square feet of heating surface are perfectly possible, and, with adequate combustion space, would be even more economical. Boilers of such a size, however, should have oil burners at each end, as in the latest German practice. The San Luis boiler referred to has an evaporative value exceeding 14 when burning 1 pound of oil per square foot of heating surface. At 12 pounds per shaft horsepower this is equal to 10,000 horsepower from one unit. Eight such boilers would produce nearly the same amount of steam as the *Lusitania*

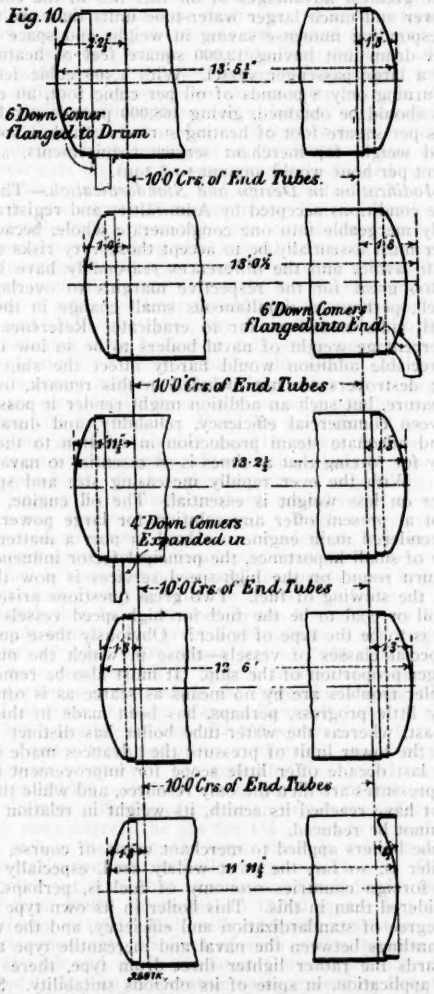
required on trial! The author would have no hesitation in building for naval work a 12,000-square foot double-ended oil-fuel unit, but other considerations, than individual full power, limit the advisability of such immense boilers.

One of the greatest advantages of oil fuel lies in the concentration of power in fewer and much larger water-tube units than are now installed, with a corresponding immense saving in weight and space adopted. Assume a three-drum unit having 12,000 square feet of heating surface to be built for a large passenger vessel. With 1,400 cubic feet combustion space, and burning only 8 pounds of oil per cubic foot, an evaporation of fifteen times should be obtained, giving 168,000 pounds per hour per unit, or 14 pounds per square foot of heating surface. Such a boiler, including water, would weigh, for merchant service requirements, about 80 tons. The fuel burnt per hour would amount to 5 tons.

Possible Modification in Design and Standardization.—The wide divergences in the conditions accepted by Admiralties and registration societies are not easily mergeable into one conglomerate whole, because the policy of the former must essentially be to accept those very risks that the latter must strive to avoid, and the differences reasonably have been, and are at present, too great for the respective margins to overlap. This is a position which, perhaps, a simultaneous small change in the engineering policy of both sides would go far to eradicate. Reference to Table II shows the percentage weight of naval boilers to be so low in some cases that an appreciable addition would hardly affect the ship as a whole. Torpedo-boat destroyers are not included in this remark, owing to their exceptional nature, but such an addition might render it possible to strike a mean between commercial efficiency, reliability, and durability on the one hand, and adequate steam production in relation to the low weight and capability for forcing that at times is so essential to naval engineering on the other. With the ever rapidly increasing size and speed of ships greater power on less weight is essential. The oil engine, whatever its future, cannot at present offer any solution for large powers. Since the turbine has rendered main engine overhaul in port a matter of rapidity, and relatively of small importance, the principal factor influencing the time required to turn round on the high-speed services is now that of boiler attention and the stowing of fuel. Two great questions arise in this connection. Is oil or coal to be the fuel for high-speed vessels? and, if the former, what is to be the type of boiler? Obviously these queries are restricted to special classes of vessels—those in which the machinery element is a larger proportion of the ship. It must also be remembered that cylindrical-boiler troubles are by no means as scarce as is often supposed. Comparatively little progress, perhaps, has been made in this design for some years past, whereas the water-tube boiler has distinct potentialities before it. At the lower limit of pressure the advances made in condenser design in the last decade offer little scope for improvement in economy; higher initial pressures are then the only resource, and while the cylindrical boiler may not have reached its zenith, its weight in relation to its steam production cannot be reduced.

Of water-tube boilers applied to merchant work, of course, the Babcock & Wilcox boiler is, so far, the most widely used, especially abroad; incidentally, in foreign countries economy of fuel is, perhaps, even more seriously considered than in this. This boiler in its own type has reached a very high degree of standardization and efficiency, and the variations in design and scantlings between the naval and mercantile type are insignificant. As regards the rather lighter three-drum type, there are few instances of its application, in spite of its obvious suitability. Several features in its design recommend it, provided that the curvature of the tubes is either small, as in the White-Forster type, or nil, as in the Yarrow type. The first modification to render the latter acceptable to registration so-

cieties will probably be in the thickness of tubes and their diameter. The German Navy adopts a standard tube of 36 mm. outside diameter and 3 mm. thick—that is, 1⅜ inches (full), and between 10 to 11 L.S.G. This is distinctly lighter than the United States 1⅜ inches tubes. The adoption



of large tubes of 1¾ inches diameter for large vessels, and small tubes of 1⅜ inches diameter for small ships, does not seem logical at first sight, nor, unless it has been purely evolutional, does the use of tubes varying 1/16 inch in diameter. There is no doubt that a tube of 1¾ inches in

diameter of 9 L.S.G. should last longer than one of 1⅛ inches in diameter of 11 L.S.G., but not necessarily longer if that tube be increased to 10 L.S.G.—that is, from 0.116 to 0.128 inch thick. A point of no little difficulty to arrive at will be the diameter of tube. To abandon the smaller sizes reduces the weight advantage and capacity for higher consumptions per unit of grate surface. The larger tubes admit of easier cleaning. It is in the excellence of the quality of the tubes, their treatment, and the workmanship lavished on their installation and preservation that the required durability of the boiler will largely depend. In view of recent naval experience, there should be no need to doubt that a design based on mercantile service requirements would prove suitable, provided that ample care was taken in regard to the quality of tubes accepted.

Merchant service fire bars, doors and casings will be required, as well as heavier brickwork, than is found in naval boilers. The drums and ends of the usual boilers will probably require thickening, and they might also be somewhat increased in size with advantage. Funnels and uptakes will remain as before. Nevertheless, in spite of this, the weight saved—of which the gain is largely in water carried—will still be very considerable. As a set-off against the additional weight, the downtake tubes outside the boiler might well be dispensed with, though builders are not in agreement on this point. It will be seen from Fig. 10 what difference in over-all length to the same boiler is made by the adoption of these tubes. It is very doubtful if any economical advantage is gained from them. For boilers with sharply-curved tubes, in which the flow of water must necessarily be less rapid than in straight tubes, they are probably necessary, but the former type is not here considered. Table IV, compiled at random

TABLE IV.

Boiler.	Heating surface.	Steam production.	Pounds steam per square foot, H.S.	Ratio of downtake area. / Generating tube area.	Type of ship and fuel.
	Sq. ft.	Lbs. per hr.			
A	3,660	28,700	7.55	4.18	Cruiser, coal.
B	4,200	31,500	7.5	4.21	Cruiser, coal.
C	3,750	25,000	6.66	3.15	Battleship, coal.
D	5,100	46,000	9.0	4.5	Destroyer, coal.
E	6,400	80,000	12.5	3.03	Destroyer, oil.
F	8,500	130,000	15.2	4.9	Destroyer, oil.

from various types, shows how little consistent relationship there is between the areas of these external pipes and of the generating tubes. That one boiler capable of producing three times the amount of steam as another should have the same size and number of downtake tubes as the lower-powered one, seems to indicate that external pipes are quite unnecessary in Yarrow-type boilers.

A further advantageous modification, which is perhaps an adaptation from condenser design, would seem to lie in a greater variation of longitudinal tube pitching between the fire rows and the outside tube rows. General practice is for the two fire rows to be of slightly greater diameter, and consequently further apart than the remainder. For instance, in a 20-row nest, the two fire rows may be 1⅜ inches in diameter, with a longitudinal pitch of 1 15/16 inches, the remaining 18 being all 1⅛-inch tubes pitched 1 11/16 inches. In the larger units that seem probable in future,

greater economy might possibly be maintained by making, in a 24-row boiler, a variation, both of tube diameter and pitch, such as 6, 8 and 10 (or 5, 7 and 12) rows, the latter being the outer ones, though obviously this would involve greater trouble in manufacture and in cleaning after use. The temperature drop, however, across a nest of water-filled tubes is so great, and the consequent volume of gas so greatly reduced, that investigation in this line would seem to be worth undertaking.

Just as existing furnaces are standardized, in view of the present amount of repetition work in boiler manufacture, so the corresponding portions of water-tube boilers can be. Upper drums, except for special cases, vary little in diameter, and need vary less. Bottom reservoirs consequently, as well as the chamber ends, could also, as far as radius and thickness, be brought to a greater state of similarity. In England and the United States top and bottom drums are made of riveted plates; in France and Germany, especially the latter, they are often made solid-drawn or welded in the circular form necessary for the bent tubes employed.

Appendix No. 1.

Of the cases of vessels whose boiler weight is relatively great in proportion to the displacement it is interesting to take two cases in order to analyze the problems involved.

Large, Fast Liners—"Lusitania" Type.—The existing arrangement comprises 23 double-ended and 2 single-ended cylindrical boilers of 17 feet 6 inches diameter, with a heating surface of 158,350 square feet and a grate surface of 4,048 square feet. The steam production, when burning about 24 pounds per square foot of grate per hour, is about 1,000,000 pounds per hour, or, say, 250 pounds per square foot per hour. The coal used for this is about 44 tons per hour, costing on an average, perhaps, 35*l.* per hour for fuel. Of the steam production, 15 per cent. is used by the auxiliary machinery. The actual ratio of evaporation, with feed at 196 degrees F., is 10.2; that is, 10.9 from and at 212 degrees F.

By adopting a combustion rate of 36 pounds per square foot of grate per hour for Yarrow boilers, and allowing for a corresponding ratio of evaporation equal to 9.5, 342 pounds of steam per square foot of grate per hour should be obtained, and for the same steam production 2,920 square feet of grate and 161,000 square feet of heating surface would be required if the ratio of heating surface to grate surface = 55.1. This would entail a coal consumption of 46.9 tons per hour at a cost of 37.5*l.* Apart from coal burnt in harbor, if the vessel is credited with 30 single trips a year, of 120 hours each, the additional cost for fuel would be—

$$2.5l. \times 30 \times 120 = 9000l. \text{ per annum}$$
$$= 300l. \text{ per trip, and the extra weight to be carried would be}$$
$$2.9 \times 30 \times 120 = 10{,}450 \text{ tons per annum}$$
$$= 348 \text{ tons per trip.}$$

At first sight these figures, due entirely to the assumed lower evaporation value of the three-drum type of boiler, appear rather formidable.

To instal 2,920 square feet of grate would require 40 boilers of 73 square feet each, though 46 of 63.5 square feet would correspond better with the existing arrangement. As a steady steaming unit burning 36 pounds per square foot of grate per hour, an area of 73 square feet is probably just about as large as could be conveniently handled under the circumstances. In this connection reference might be made to the double-ended boiler of this type which has been adopted in a few torpedo-boat destroyers, but

in no larger vessels, and is not considered here, although it is hard to see why, if a single-ended boiler will do, a double-ended one should not equally be adopted, with the consequent avoidance of one of the principal objections to the single-ended type—that is, multiplicity of units. When the additional steam and feed piping, uptake connections, etc., are considered, together with the greater supervision necessary, it must be admitted that the increased number of units introduces a distinct objection to the adoption of water-tube boilers, unless large units are accepted.

Again, the *Lusitania's* double-ended boilers are 22 feet long. The length of the boilers proposed would not be less than 12 feet each, with, say, 1 foot 3 inches between them, so that they would require an additional frame space in each stokehold—say, 32 inches, because it would not be advisable to reduce the existing length of firing space in a ship of such size. The objection to increased length rather emphasizes the desirability of double-ended boilers, and the adoption of the water-tube type is in itself a sufficiently large step to render these a relatively less risky additional experiment. The number of units would then be the same as at present, except for the two double-ended and two single-ended boilers forward. Assuming, however, the adoption of forty single-ended boilers, the weight of generators only, even with increased scantlings, would amount to about 1,800 tons, a saving probably of 2,200 tons on the existing design, against which is set the additional 350 tons of coal, reducing the total to a net amount of about 1,800 tons. The lower weight of seatings may be assumed to counterbalance the increased weight of connections.

There are then the following advantages:

1. A slight saving in fore-and-aft space of about eight frame spaces; or
2. A gain of 1,800 tons of cargo at, say, 10s. per ton per trip; or
3. A vessel of 34,200 tons, against 36,000 tons; or
4. A draught of water reduced from 32 feet 9 inches by at least 12 inches; or
5. A slightly finer ship, together with rather less power required, or, *vice versa*, higher speed.

It is assumed that it would be possible to stow the extra 350 tons of coal with very slight rearrangement of bulkheads, because the over-all width of the boilers proposed would only be 16 feet 6 inches, against the existing 17 feet 6 inches, and that this, in conjunction with any cargo carried, would probably sufficiently compensate for the altered center of gravity of machinery.

If only half the available cargo space were filled, the additional revenue would be, say, $900 \times 30 \times 0.5l. = 13{,}500l.$, reducing the additional fuel bill from a loss of 9,000*l.* to a gain of 4,000*l.* But it is, in the author's opinion, still doubtful if the case of Yarrow or other water-tube boilers is not understated.

In the event of double-ended boilers being adopted, the efficiency would probably be even higher, and the weight saved rather more. In any case, apart from any practical disadvantages—which, by the way, are probably more anticipated than actual, as modern naval practice goes far to prove—the influence of reduced weight on a ship as a whole is so great as to render it difficult to form an absolutely definite opinion without most careful investigation of the details involved. The attraction of such a saving is very tempting.

The use of oil fuel should not be overlooked in conjunction with the proposed boilers, as with the large units possible the combined influence on the design of ship is enormous. With oil fuel the stokehold complement would at once be reduced to one-fifth of the present number, and the simplification in stokehold design should go far to balance the higher cost of water-tube boilers. If the arrangement of the *Lusitania's* three after stokeholds were retained, each containing six boilers, each of the eighteen would require to supply about 56,000 pounds of steam per hour. This

would mean relatively small units, and twelve boilers providing 84,000 pounds apiece would be more suitable. An evaporation of fifteen times at 200 degrees F. feed temperature should be obtained, requiring 5,600 pounds of oil per boiler per hour. By burning 8 pounds per cubic foot, units of 7,000 square feet of heating surface and 700 cubic feet of combustion space would suffice. This should enable the vessel to be built with two funnels instead of four, at a large gain of passenger accommodation, and to carry only 60 per cent. of the fuel now required. With the enormous gain in weight of boilers and fuel, together with the smaller space occupied, any additional cost in maintenance, or doubt as to regularity of supply, must assume an aspect of relatively small amount.

Channel Steamers.—In the case of Channel steamers the considerations are very different. The longest run is barely four hours; economy is of relatively less importance, the dimensions of the ship are usually restricted, and the proportionate weight of cylindrical boilers is generally very high.

Several 23-knot Channel boats of about 2,700 tons and 12,000 shaft horsepower have recently been built, carrying eight single-ended boilers of 180 pounds pressure, which weigh, say, 65 tons apiece, making together very nearly 20 per cent. of the total displacement of the ship. When burning 36 pounds per square foot of grate surface per hour on 465 square feet, the evaporation is 9.6 times from feed at 120 degrees F.—that is, 11.06 times from and at 212 degrees F. For Yarrow boilers, burning 40 pounds per square foot, an evaporation of 10.5 times is assumed, and is equal to 9.15 times at 120 degrees F. In the former case the steam produced per square foot of grate is about 350 pounds per hour; in the latter 366 pounds, necessitating 440 square feet of grate surface. Six boilers, each with 73 square feet of grate surface and 3,800 square feet of heating surface, would be sufficient. Increased to Board of Trade scantlings they would weigh about 33 tons apiece with water—say 200 tons in all, or a gain of over 300 tons, which is more than 10 per cent. of the displacement.

The reduction in power possible would be

$$12{,}000 \times \frac{2{,}400^{\frac{2}{3}}}{2{,}700^{\frac{2}{3}}} = 11{,}050,$$

which, again, would admit of a further reduction in weight of boilers of 20 tons.

The vessel is 350 feet by 41 feet by 13 feet draught. As in the larger ship, the saving can be applied in many ways. The alteration to center of machinery weight is more serious in this case, and the engines and boilers would require to be moved bodily forward.

If the saving be used for additional cargo, this matter is of less importance; if it be used to fine and lighthen the vessel, an obvious saving in fuel burnt must result. Fore-and-aft space will be saved to the extent of six frame spaces, rendering a rather smaller ship possible.

If oil fuel is burnt in place of coal, the number of boilers would be reduced to four, and the heating surface also reduced considerably; in fact, to produce 12 pounds of steam per square foot of heating surface from feed at 120 degrees F. would require four boilers of less than 3,500 square feet each. In this case the gain in the ship, as a whole, would be very great indeed. Instead of two funnels only one need be fitted. The size and initial cost of the vessel are reduced. The total fuel burnt per hour in the original design at full power is about 7.5 tons, which would be equal to 5.5 tons of oil, or, say, rather under 5 tons with the reduction probable in the ship.

The requirements for such a service would be such that no difficulty should be experienced in obtaining a regular supply of oil.

Appendix II.

Oil Fuel.—Two-thirds of the present world's oil supply comes from the United States, the majority being Californian; Russia supplies 24 per cent., the remaining 10 per cent. being very widely distributed over about fifteen different countries. The total production of oil in 1905—the year when oil fuel alone was adopted for certain of His Majesty's ships—was about 37,000,000 tons; in 1911 it was nearly 53,000,000 tons. For United States oil fuel, as used in Europe, a calorific value of 19,400 British thermal units per pound is a fair average, giving an equivalent evaporation from and at 212 degrees F. of 20.1 pounds of water. The cost in England is now about one-half of what it was five years ago. The tendency for the sources of supply to increase is still accompanied by a slight reduction in cost per ton.

The two principal objections to the use of oil fuel are: (1) Uncertainty of supply. The United States are in a very strong position in regard to oil fuel, and the decision of the United States Navy Department to instal an oil-burning system only in their last two battleships is very significant;

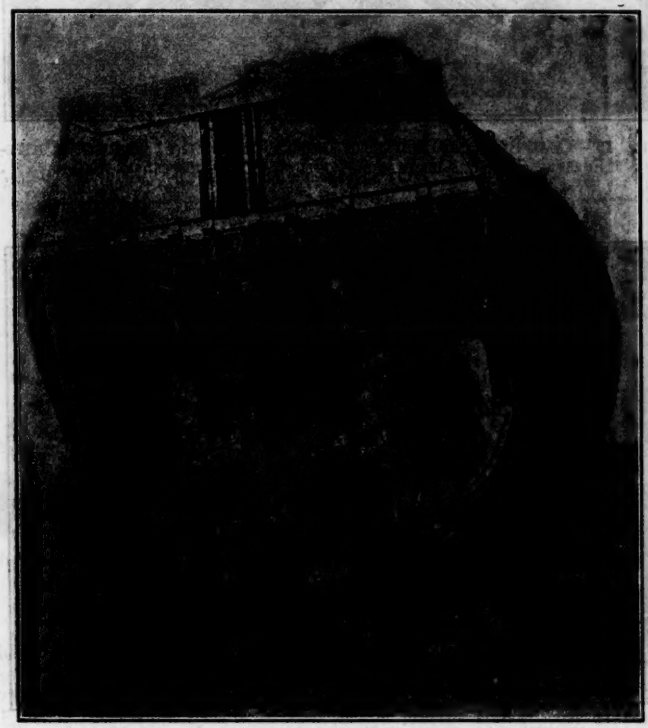


Fig. 11.—Single-Ended Boiler, 17 Feet 6 Inches in Diameter, 215 Pounds Pressure.

FIG. 12.—COAL-BURNING BOILER OF 110 SQUARE FEET GRATE (WHITE-FORSTER TYPE): GENERATOR PORTION ONLY ERECTED IN SHOP.

FIG. 13.—OIL-BURNING BOILER, 8,500 SQUARE FEET OF HEATING SURFACE; GENERATOR PORTION ONLY ERECTED IN SHOP.

the potential advantages of doing so are greater for them than for anyone else. Oil-burning European liners would require to take in fuel on the American side, and the certainty of being able to do so is, perhaps, rather doubtful, compared with the United States Navy supply. A number of large, fast liners burning oil would make a serious inroad into the world's production as it stands at present. For instance, ten large Atlantic liners, each burning 40 tons per hour on fifteen round trips a year of 120 hours each way, would require 3 per cent. of the world's present output. The question of the site and the cost of storage plant in close proximity to the liner's docks are other matters of considerable difficulty.

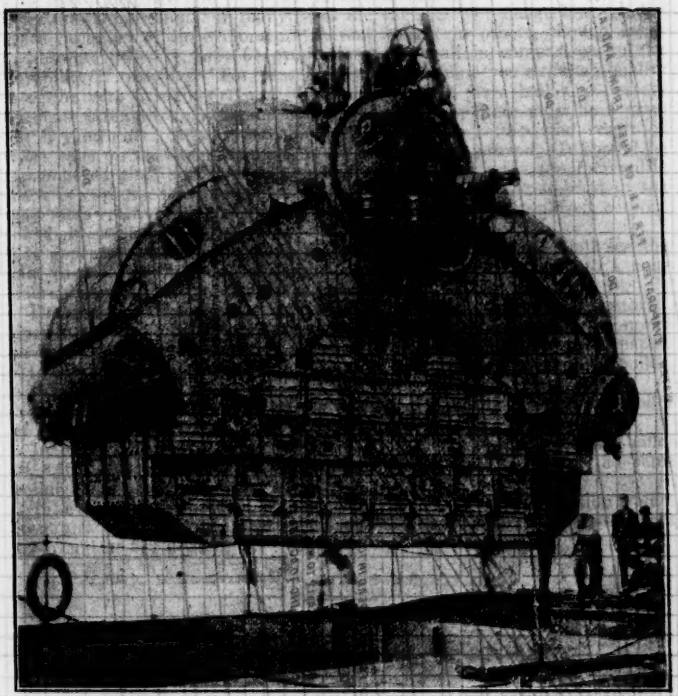
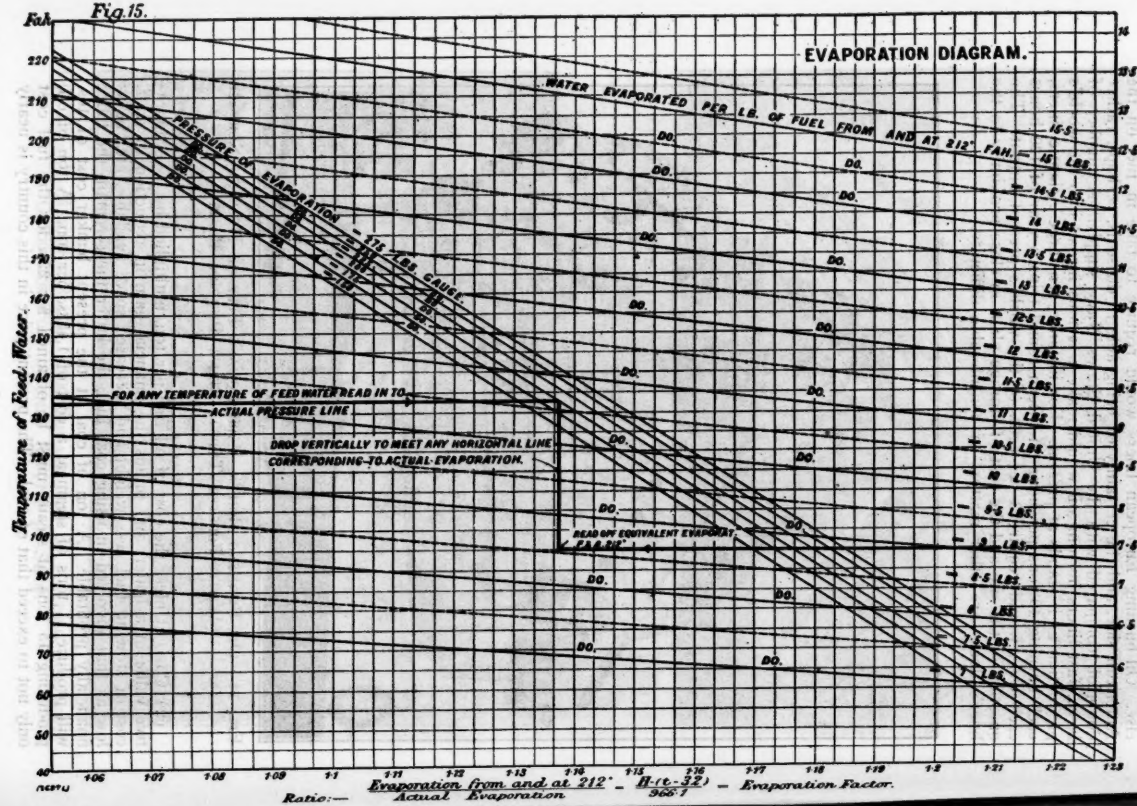


FIG. 14.—OIL-BURNING BOILER, 8,500 SQUARE FEET OF HEATING SURFACE (WHITE-FORSTER TYPE), BEING PUT ON BOARD.

(2) Cost. Unless the cost of oil fuel per ton, multiplied by its evaporative value in a boiler, and minus the financial gain in reduction of staff, cost of stowing fuel, less weight and increased space available, etc., be as low as that involved in using coal fuel, the latter must remain the more financially profitable. A ton of coal at the present bunker cost of 16s. will produce 11 tons of steam at a cost of 1.453s. per ton. A ton of oil, producing 15 tons of steam, must be obtained for 22s. for the fuel cost only not to exceed that of coal. The present price in this country is nearly

Fig. 15.


EVAPORATION DIAGRAM.
WATER EVAPORATED PER LB. OF FUEL FROM AND AT 212° FAH. — 15 LBS.
DO.
PRESSURE OF EVAPORATION — LBS. GAUGE.
FOR ANY TEMPERATURE OF FEED WATER READ IN TO ACTUAL PRESSURE LINE
DROP VERTICALLY TO MEET ANY HORIZONTAL LINE CORRESPONDING TO ACTUAL EVAPORATION.
READ OFF EQUIVALENT EVAPORAT. F.A.B. 212°
Temperature of Feed Water.
Actual Evaporation in Lbs. Water per Lb. of Fuel under Trial Conditions of Feed Temperature and Steam Pressure.
Ratio:— Evaporation from and at 212° / Actual Evaporation = H-(t-32) / 965.7 = Evaporation Factor.

NOTES.

double this figure, and even if obtained in large quantities for as low a price as 30s. per ton, the oil fuel cost would exceed that of coal by 33 per cent. This difference, however, would probably be made up in the advantages of reduced staff, etc., as described above.

Summary.—There is little doubt that the present reputation of the water-tube boiler as an efficient and durable steam generator has suffered from its early history. It has become so much a matter of course to instal cylindrical boilers in fast ships that the water-tube type has been largely ignored by ship owners, and when suggested by builders has been ruthlessly suppressed by survey conditions. Many practical questions of steam-ship operation that find no place in considerations of thermal efficiency enter into ship design. In many cases the vagaries of tonnage laws render space occupied of little moment. Actual weight in others, though large in absolute measure, may be proportionately small in comparison to the durability it provides. For passenger and merchant vessels, that quality is of the very first importance. It is not sufficient for a boiler to possess adequate strength to withstand ordinary conditions; there must be a margin beyond this. Again, the ability to realize on service the result of short runs on trial trips remains to be more widely proved. It should be remembered that no advocacy of high combustion rates for mercantile work is expressed in the paper, although for oil boilers it is hard to see why rates relatively high to those of coal should not be employed without the slightest danger.

It appears extremely probable that in the near future we shall see the gain to the ship in reduced weight of boilers and fuel valued to a considerably greater extent than is at present the case.

The question of superheat from water-tube boilers is not to be ignored. Even a moderate amount has a most marked effect on engine economy on board ship, but the question is too long to discuss in the present paper.

In Fig. 11 is shown the front view of an ocean-going vessel's boiler, with Howden draft fittings.

In Fig. 12 is shown the generator portion of a White-Forster coal boiler of 110 square feet of grate surface.

In Fig. 13 is shown the generator portion of a White-Forster oil boiler of 8,500 square feet of heating surface; while Fig. 14 shows the same boiler being placed on board the vessel.

In Fig. 15 is a diagram by means of which actual evaporation values from any feed temperature may be rapidly converted into equivalent values from and at 212 degrees F.—"Engineering."

DETERIORATION OF COAL IN STORAGE.

AMOUNT DETERMINED FROM ACCURATE TESTS BY THE BUREAU OF MINES.

H. C. Porter and F. K. Ovitz—American Chemical Society.

Until very recent years coal has been considered as an extremely unstable material, subject to serious alteration and losses when exposed to the weather. Even as late as 1906 a paper read before the United States Naval Institute states: "The pressure of the weight of coal causes gases to be evolved; . . . these gases constitute the chief and only value of the coal in that they furnish the heat units. It is claimed that if a ton of fine bituminous coal be spread out on a concrete pavement . . . in the open air in this climate (Key West, Florida) for one year, it will lose all its calorific properties. The gases are simply free to escape, and when

the coal has lost all its gas it will have lost all its heat units and be simply coke."

Other statements, even more emphatic, have appeared from time to time, and while probably most chemists of today do not hold such extreme views, yet there remains in the minds of most an impression that deterioration does occur to such an extent as to be of industrial importance. For this reason the Bureau of Mines undertook to determine accurately the extent of deterioration in different types of coal.

First, a study was made in the laboratory of the loss of volatile matter from crushed coal during storage. A number of samples (20 pounds each) representing a variety of types from widely separated fields, were broken to about ½-inch size and immediately placed in glass bottles in the mine. At the laboratory the accumulated gas was withdrawn and a free continuous escape of the volatile products permitted at atmospheric pressure and temperature. The results of these experiments show that while several coals evolve methane in large volumes, especially in the early period after mining, the coal suffered in one year a loss of calorific value from this cause of but 0.16 per cent. as a maximum.

It seems, therefore, that the loss due to escape of volatile matter from coal has been greatly overestimated.

At the instance of the Navy Department, however, more elaborate tests were undertaken to determine the total loss possible in high-grade coal by weathering. The extent of the saving to be accomplished by water submergence as compared to open-air storage was a point to be settled, and there has also arisen the question as to whether salt water possessed any peculiar advantage over fresh water for this purpose.

In brief outline the tests by the Bureau of Mines were carried out as follows: Four kinds of coal were chosen: New River on account of its large use by the Navy, Pocahontas as a widely used steaming and coking coal in the eastern section, and as being also the principal fuel used in the Panama Canal work, Pittsburgh coal as a type of rich coking and gas coal, and Sheridan (Wyo.) sub-bituminous or "black lignite"—a type much used in the West. With the New River coal, fifty-pound portions were made up out of one large lot, which had been crushed to one-half inch size and well mixed. These portions, confined in perforated wooden boxes, were submerged under sea water at three navy yards, differing widely from each other in climatic conditions, and 300-pound portions from the same original lot were exposed to the open air, both out of doors and indoors, at the same places.

With the Pocahontas coal, test was made only at the Isthmus of Panama, run-of-mine coal being placed in a 120-ton pile, exposed to the weather. Pittsburgh coal was stored as run-of-mine in open outdoor bins, five tons capacity, at Ann Arbor, Mich., also in 300-pound barrels submerged under fresh water. The Wyoming sub-bituminous was stored at Sheridan, both as run-of-mine and slack in outdoor bins, holding three to six tons each.

Every test portion was sampled each time in duplicate, and in all cases, except the outdoor pile at Panama and the 300-pound open-air piles of New River coal, the sampling was done by rehandling the entire amount.

Small lots and a fine state of division were conditions purposely adopted with the New River coal so as to make the tests of maximum severity.

Moisture, ash, sulphur and calorific value determinations were made on each sample, the last by means of the Mahler bomb calorimeter and a carefully calibrated Beckman thermometer. The calorimetric work on all except the Sheridan, Wyo., tests has been done throughout by one man, Mr. Ovitz, and with the same instrument. All the calorific values have been calculated to a comparable unit basis, viz: that of the actual coal substance free of moisture, sulphur and corrected ash.

The results show in the case of the New River coal less than 1 per

cent. loss of calorific value in one year by weathering in the open. There was practically no loss at all in the submerged samples, and fresh water seemed to "preserve the virtues" of the coal as well as salt. There was almost no slacking of lump in the run-of-mine samples, and the crushed coal in all cases deteriorated more rapidly than run-of-mine.

The Pocahontas run-of-mine in a 120-ton pile on the Isthmus of Panama lost during one year's outdoor weathering less than 0.4 per cent. in heating value, and suffered little or no physical deterioration of lumps.

The Pittsburgh gas coal during six months outdoor exposure suffered no loss whatever of calorific value, measurable by the calorimetric method used, not even in the upper surface layer of the bins.

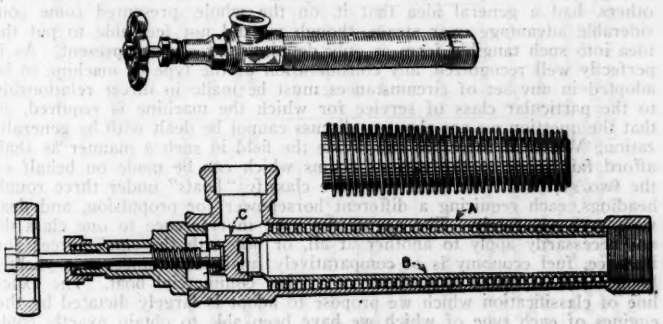
The Wyoming coal lost as much as 5.3 per cent. in one of the bins during 2¾ years, and 3.5 per cent. even in the first three months. There was bad slacking and crumbling of the lumps on the surface of the piles, but where the surface was fully exposed to the weather this slacking did not penetrate more than 12 to 18 inches in the 2¾-year period.

Storage under water unquestionably preserves the heating value and the physical strength of coal. But it practically necessitates firing wet coal, and therefore means the evaporating in the furnace of an amount of moisture varying from 1 to 15 per cent., according to the kind of coal. This factor is an important drawback to underwater storage with coals like the Illinois and Wyoming types, which mechanically retain 5 to 15 per cent. of water after draining, but in case of the high-grade eastern coals, if firemen are permitted, as is ordinarily the case, to wet down their coal before firing, "so as to make a hotter fire," as they say, then the addition during storage of the 2 or 3 per cent. moisture which these coals retain would be of little consequence. Submergence storage is an absolute preventative of spontaneous combustion, and on that account alone its use may be justified with some coals, but merely for the sake of the saving to be secured by avoidance of weathering there is no ground for its use.—"The Engineering Magazine."

A SUBSTITUTE FOR A TRAP.

The accompanying figure shows the design of a device intended to handle water of condensation and which has been given the name "Ratisal" by its inventor, Heinrich C. Sommer, Nachfolger, Düsseldorf, Germany.

A tube-like conical shell *A* contains a removable tube *B* upon the external surface of which is formed a spiral screw as shown. During nor-



DETAILS OF CONSTRUCTION

mal operating conditions the valve *C* is closed and the water of condensation is forced to find an exit through the sinuous canal of gradually decreasing cross-section formed by the spiral screw and the inner surface of the shell *A*.

As the canal is of considerable length, the frictional resistance offered to the passage of the water is almost equal to the pressure of the steam in the space being drained. By adjusting the sectional area of the canal the frictional resistance may be regulated so that the water will have any desired pressure as it emerges from the device up to within a small degree of that of the steam. For this purpose two sizes of copper-wire spirals are supplied which may easily be wound into the screw to diminish the effective area.

If air is to be blown out of the system which the device is draining or if for any reason a temporary vigorous steam or water discharge is desired, the valve *C* is opened and a short-circuit is established through the central core of the tube *B*.—"Power."

AN ANALYSIS OF THE CLAIMS OF THE MARINE INTERNAL-COMBUSTION ENGINE.

Many enthusiastic partisans of the internal-combustion engine for use afloat have comparatively hazy notions as to what exactly are the advantages both in nature and extent which it has over the steam engine, and it is to be feared that they are prepared, in some instances at all events, to make rather exaggerated claims for it. On the other hand, there are supporters of the older form of power who are not prepared to admit more than a trifling, even if any, advantage in any direction over steam. This is hardly to be wondered at, seeing that reliable comparative data are very difficult to come by, and require analyzing with an unbiased mind in the light of some knowledge of both forms of power in actual practice. Moreover, any data now available are still largely founded on the estimate, or perhaps guarantees, of the manufacturer, based upon his past experience, somewhat limited as that must be. Such data we have been trying to collect for some time, and we propose to endeavor to put a few facts forward in such a way as shall present an absolutely unprejudiced statement of the case, and so try to arrive at some approximately accurate idea as to how the question really does stand. We hold no brief for either form of power, though we must admit that before taking up the study of the marine oil engine we, in common with many others, had a general idea that it, on the whole, presented some considerable advantage over steam, though we did not feel able to put the idea into such tangible form as only definite figures can represent. As is perfectly well recognized, any consideration of the type of machine to be adopted in any set of circumstances must be made in direct relationship to the particular class of service for which the machine is required, so that the question we are about to discuss cannot be dealt with by generalization. We must therefore sub-divide the field in such a manner as shall afford fair scope to the various claims which can be made on behalf of the two systems. We shall therefore classify "boats" under three rough headings, each requiring a different horsepower for propulsion, and deal with each separately. The arguments used in reference to one class do not necessarily apply to another at all, or not with the same force; for instance, fuel economy is of comparatively small importance on a yacht's launch, but of prime importance in a large commercial boat. The exact line of classification which we propose to adopt is largely dictated by the engines of each type of which we have been able to obtain exactly com-

An Analysis of the Claims of the Marine Internal-Combustion Engine.

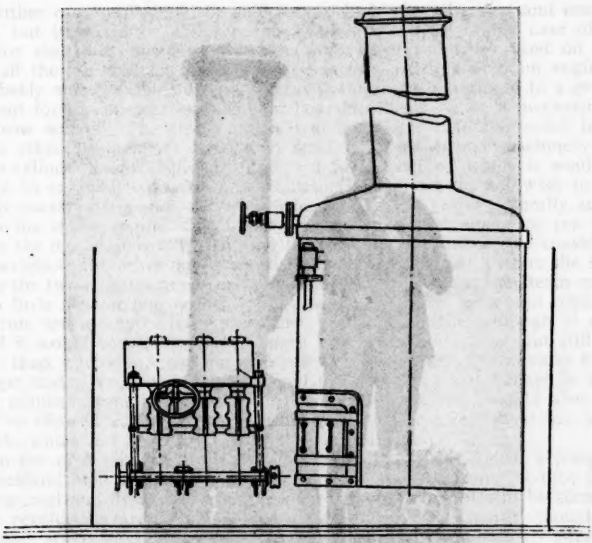
Name.		Cost.	B.H.P.	Revs.	Weight.	Fuel used per H.P. per hour.	Fuel used.	Fuel cost per 24 hours.	Wages.	Length.	Remarks.
		£			cwt.			£ s. d.		ft. ins.	
Simpson, Strickland & Co.	Steam	235[6]	14.4[9]	750	11[1]	1.5 lbs. per I.H.P.	Paraffin at 4d. per gal.	1 5 2	.....	4 4½	Length of engine and boiler.
										4 10½	Length between bulkheads.
Simpson, Strickland & Co.	Steam	235[6]	14.4	750	11[1]	3 lbs. per I.H.P.	Coal at 17s. 6d. per ton.	0 9 5	.....	5 4½	Length of engine and boiler.
										6 0	Length between bulkheads.
Parsons' Motor Company.	Oil	146[6]	14	750	8¼[4]	0.75 pint per B.H.P.	Petrol at 1s. per gal.	1 11 6	.....	4 10½	Length of engine over all.
Parsons' Motor Company.	Oil	146[6]	14	750	8¼[6]	0.75 pint per B.H.P.	Paraffin at 4d. per gal.	0 10 6	.....	6 0	Length between bulkheads. Add something for petrol for starting up.
					tons.						
Simpson, Strickland & Co.	Steam	450[7]	80	320	8½[8]	3 lbs. per I.H.P.	Coal at 17s. 6d. per ton.	2 12 10	1 man.	14 10	Length of engine and boiler.
										16 0	Length between bulkheads.
Griffin Engineering Company (reversing gear).	Oil	550[7]	80	350	5[5]	0.625 pint per B.H.P.	Fuel oil at 45s. per ton.	1 4 9	1 man.	13 8½	Length of engine.
										16 0	Length between bulkheads.
Blackstone & Co. (reversing engine).	Oil	420[7]	80	450	3.4	0.6 lb. per B.H.P.	Fuel oil at 45s. per ton.	1 3 1	1 man.	8 0[10]	Length of engine only.
					tons.	Fuel used per 24 hours.					
David Rowan & Co.....	Steam	5,600[2]	1,100	130	186[2]	21 tons.	Coal at 17s. 6d. per ton.	18 7 6	3 firemen, 3 greasers, 3 engineers } £54 per month	40 0	Length between bulkheads.
Nederlandsche Fabriek (4-cycle).	Oil	13,000[2]	1,100	130	190[2]	5.5 tons.	Fuel oil at 45s. per ton.	12 7 6	3 greasers, 3 engineers } £42 per month	26 0	Length of engine over all.
										33 0	Length between bulkheads.
M. A. N. (2-cycle, double-acting).	Oil	8,100[2]	1,200	130	90[2]	6.1 tons.	Fuel oil at 45s. per ton.	13 14 6	Ditto.	31 6	Length between bulkheads.
										38 0	Length of engine over all.
Two-cycle, single-acting[11]	Oil	8,200[2]	1,100	130	96	.....	.....	...	.....	...	.....

1. Weight with steam up, and all auxiliaries and stern gear. 2. Cost fitted on board, with engine-room auxiliaries down to thrust block. 3. Weight with all auxiliaries and down to crank-shaft coupling. 4. Weight with stern gear and fuel. 5. Weight of complete engine-room plant, with steam up in the case of the steam engine. 6. Weight of engine, boilers and auxiliaries with steam up, and down to crank-shaft coupling. 7. Cost of complete plant ready to fit on board, engine-room outfit only. 8. List price. 9. Sold as 18 I.H.P. at 800 revolutions per minute. 10. Approximately. 11. Weight and cost based upon an engine of 1,200 B.H.P. at 130 revolutions per minute.

parable data, and, though it may not be considered entirely satisfactory, we think that, on the whole, the divisions are fair, each one meeting the next, and the whole covering practically all the ground.

In making any such comparison as that proposed it is, first of all, necessary to arrive at some common denominator to which all the engines in each class can be reduced, so that none can obtain an unfair advantage over another by reason of the omission of some factor from the calculation, and we think that that common basis is best found by basing all the comparisons upon an equal number of revolutions; by this means all question of propeller efficiency and hull form are eliminated, and speed need not be considered. It is, for instance, obviously unfair to take a high-speed motor and compare it with an old slow-going compound steam engine, as has been done so often; we have even seen a sketch showing the great saving in length obtained by fitting a fast-running motor into the place of a twenty-year-old steam engine with a locomotive boiler.

With this basis agreed upon and limited by the above-mentioned difficulty of getting facts as to exactly comparable engines, we find that the grouping comes out automatically as follows: (1) 14 brake horsepower, launches; (2) 80 brake horsepower, barges, small tugs, fishing smacks, etc.; (3) 1,100 brake horsepower, small tramps. This sub-division covers most types of small craft, where special conditions do not apply, and may, with due discretion, be somewhat extended to types not mentioned; but we do not profess to be able to cover every possible one of the multitudinous conditions of service which exist, to some of which our arguments might conceivably apply in the inverse ratio of that given. We have taken the brake horsepower of the steam engines as 85 per cent. of the indicated horsepower in each case, which will be accounted a very fair assumption. A point which has to be borne in mind in putting forward such data is that all the engines compared shall be as nearly as possible of the same class, that is, specially designed for the same purposes and with the same quality of workmanship and material put into their construction. For the small type of steam engine we have therefore taken the 18 indicated horsepower triple-expansion launch engine with oil-fired water-tube boiler, built by Messrs. Simpson Strickland & Co. specially for launch use, and contrasted it with the 14 brake horsepower two-cylinder Parsons launch motor with reversing gear. Referring to the table, it will be seen that the brake horsepower of the steam engine is slightly more than that of the motor, but not enough to invalidate the figures, and the revolutions are exactly the same; the total weight of the steam engine with steam up is seen to be 2½ cwt., or about one-third greater than the petrol engine, and with fuel for the same number of hours steaming will be a little higher still. This weight, however, on a 30 cwt. yacht's launch will hardly be appreciable, while for a boat which is not intended for hoisting in davits it will be of no importance at all. The cost of the steam engine is, however, very much higher than the particular example of motor taken, though the prices of the latter vary considerably, and some makes are much higher than the Parsons. With regard to the space occupied, it will be seen from Figs. 1 and 2 that the steam engine is actually shorter than the motor, while the latter requires some further allowance for starting up, though this need not be considered when once under way, unless a bulkhead is required in the boat, when the loss of room would be permanent. Figs. 3 and 4 show illustrations of the actual engines compared. Coming to the question of comparative economy, it will be seen that the steam engine burning paraffin at 4d. per gallon will cost £1 5s. 2d. for twenty-four hours running, while the motor will cost £1 11s. 6d. running on petrol at 1s. per gallon for the same period, a difference which, though about 20 per cent., is not of very great importance, in view of the total number of hours such a boat is likely to be run in a year, and the small proportion the fuel costs bear to the total costs of running where a paid hand is kept.



Length over engine and boiler, 4 feet 4½ inches.

FIG. 1—19-H.P. STEAM ENGINE AND BOILER.

If, however, coal at 17s. 6d. per ton were to be used as fuel, assuming the excessive consumption of 3 pounds per indicated horsepower per hour for a small triple engine, the cost would be reduced to 9s. 5d. for the twenty-four hours, but this would introduce dirt, and more room would be required for firing, bringing the length between bulkheads up to the same as that for the paraffin engine, while the motor might counter by using paraffin at 4d. per gallon, when the cost for the twenty-four hours would come down to 10s. 6d., with a small addition for petrol for starting up, and the two would be more nearly equal under this head. No definite statement can be made as to the relative labor costs, as it would vary in different circumstances; in a yacht's launch a hand would be taken

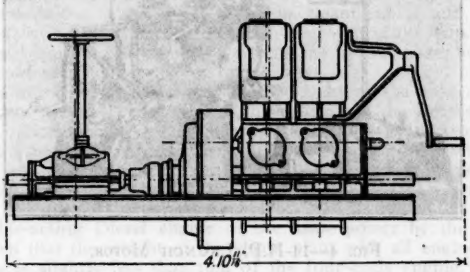


FIG. 2—14-H.P. LAUNCH MOTOR.

FIG. 3—18-H.P. STEAM ENGINE AND BOILER.

FIG. 4—14-H.P. LAUNCH MOTOR.

in either case, and would be sufficient to look after the boat and machinery, but it would be easier to dispense with a man in the case of the motor than with the steam launch, even when the latter used oil fuel. In all the sizes which are considered in this article the steam engine is probably more flexible; that is, the revolutions can be reduced to a greater extent for the purposes of easing a boat in a head sea or maneuvering in narrow waters. The steam engine will be quieter than the motor in the case taken, though we have seen small triple-expansion machinery and four-cylinder motors specially designed for silence of which it would be hard to say which was the most quiet. Though we do not wish to deal with matters of opinion in this article, yet, as it is pretty generally agreed that the steam engine is still today more to be relied upon for sea work than the motor, it is only fair that this point should at least be considered in weighing the other points mentioned. Granting, then, that in the small size the two systems are equally durable, it is found that the steam engine is a little heavier and is more costly to buy, but it is safer and costs less to run, and occupies less space than the petrol engine; though if using coal it would occupy the same length between bulkheads as, but still cost less than, a paraffin engine to run, and would take, say, five minutes longer to get under way; steam being raised in the type of boiler taken in about six minutes from all cold. A small sum—say, £2 10s.—would also have to be allowed each year for retubing the boiler after about six years' work, which has no comparable figure in the motor.

In the next size, 80 brake horsepower, we have also taken a Simpson-Strickland built steam engine; this time a compound with fire-tube boiler using coal and designed for rough, hard work. Here it will be seen that the revolutions are a little lower than those of the motor, though not enough so to affect propeller efficiency, and hardly enough to affect the weight, which is some 60 per cent. more than the motor, while the weight of fuel for equal horsepower hour steaming will be about five times as great. The first cost of the steam engine is 20 per cent. lower, but the cost of running is more than double that of the motor. The motor taken in this case is the 80 brake horsepower four-cylinder Griffin engine, which we described in our issue of November 11th, 1910. This engine is fitted with a reversing gear, and it will probably come as a great surprise to many that the space required between bulkheads is nearly the same as that required by the steam set, as shown in Figs. 5 and 6. The great importance of having all the engines to be compared running at the same number of revolutions can be gaged by observing the saving in weight, space, and cost to be obtained by increasing the revolutions from 350 per minute to 450 per minute, as seen in the table in connection with the Blackstone reversing engine. Thus it will be seen that for the same radius of action the motor-driven boat can carry about 13½ tons more cargo on the same displacement at the same speed, allowing for the difference in weight between the Griffin engine and the steam engine and the reduced quantity of fuel in the bunkers required for a four days' trip. The number of men required will probably be the same in each case, but the steam engine would take longer to get under way.

Some rather surprising figures come to light in the case of the third series of comparisons. For these we have obtained the figures of a standard 1,300 indicated horsepower return-tube boiler, triple-expansion steam equipment, from Messrs. David Rowan & Co., the well-known builders of Glasgow, and a six-cylinder four-cycle 1,100 brake horsepower Diesel engine now under construction by the Nederlandsche Fabriek, and to make the comparison complete we have also got the figures for a two-cycle double-acting Diesel engine of the same power by the M.A.N. It will be seen that the weight of the steam plant with all engine auxiliaries and piping is slightly less than that of the four-cycle engine, but there is

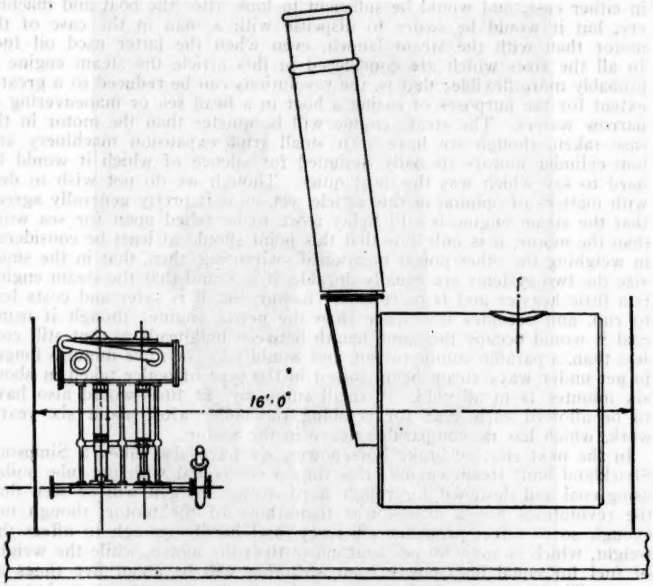


FIG. 5—80-B.H.P. STEAM ENGINE AND BOILER.

a saving of nearly 100 tons by the adoption of the two-cycle double-acting engine.

We have been unable to obtain the corresponding figures for a single-acting engine of exactly the same power, and we have therefore given figures of cost and weight calculated from horsepower from a 1,200 B.H.P. set. It will come as a surprise to note from these what a very lit-

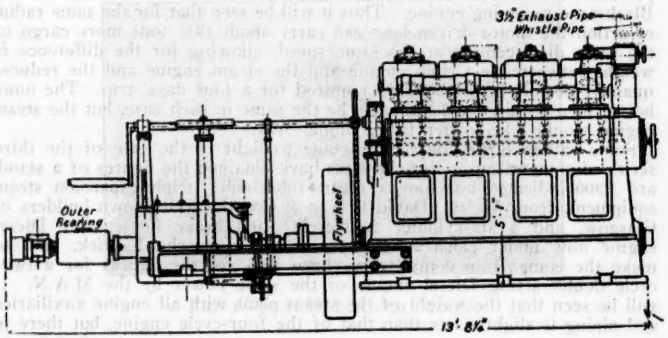


FIG. 6—80-B.H.P. OIL ENGINE.

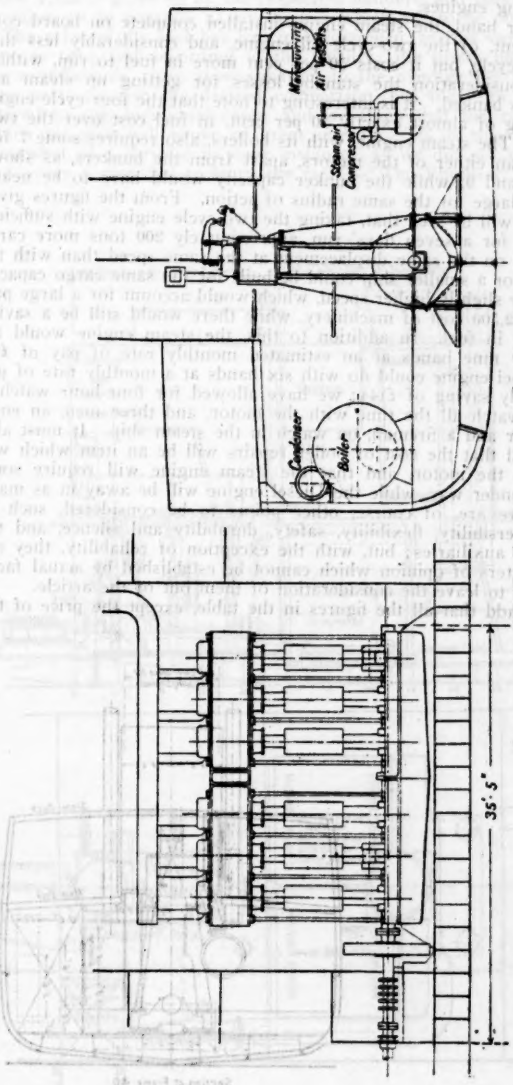


FIG. 7—1,100-B.H.P. FOUR-CYCLE OIL ENGINE.

tle difference there is in weight and cost between the double-acting and the single-acting engines.

On the other hand, the steam engine installed complete on board costs only 70 per cent. of the two-cycle oil engine, and considerably less than half the four-cycle, but it costs 50 per cent more in fuel to run, without taking into consideration the standby losses for getting up steam and lying with fires banked. It is interesting to note that the four-cycle engine shows a saving of almost exactly 10 per cent. in fuel cost over the two-cycle engine. The steam engine, with its boilers, also requires some 7 feet more space than either of the motors, apart from the bunkers, as shown in Figs. 7, 8 and 9, while the bunker capacity would have to be nearly four times as large for the same radius of action. From the figures given in the table it will be seen that, taking the two-cycle engine with sufficient fuel on board for a seven days' run, approximately 200 tons more cargo can be carried on the same displacement at the same speed than with the steam engine, or a smaller ship could be built for the same cargo capacity and a probably slightly higher speed, which would account for a large part of the extra £2,500 cost of machinery, while there would still be a saving of £6 per day in fuel. In addition to this, the steam engine would require probably nine hands at an estimated monthly rate of pay of £54, while the Diesel engine could do with six hands at a monthly rate of pay of £42, a yearly saving of £144; we have allowed for four-hour watches, two men on watch all the time with the motor, and three men, an engineer, a greaser and a fireman, on watch in the steam ship. It must also be remembered that the cost of boiler repairs will be an item which will not appear in the motor, and that the steam engine will require some hours to get under way, while the Diesel engine will be away in as many minutes. There are, of course, other points to be considered, such as reliability, reversibility, flexibility, safety, durability and silence, and the driving of the auxiliaries; but, with the exception of reliability, they are so largely matters of opinion which cannot be established by actual facts, that we prefer to leave the consideration of them out of the article.

We should add that all the figures in the table, except the price of the

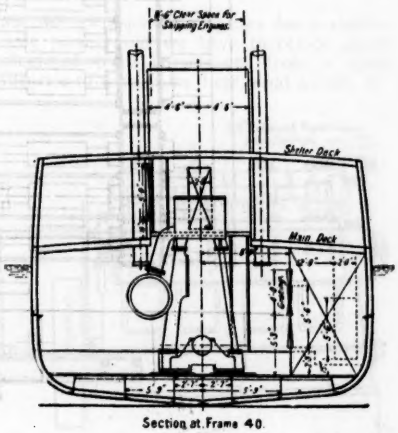
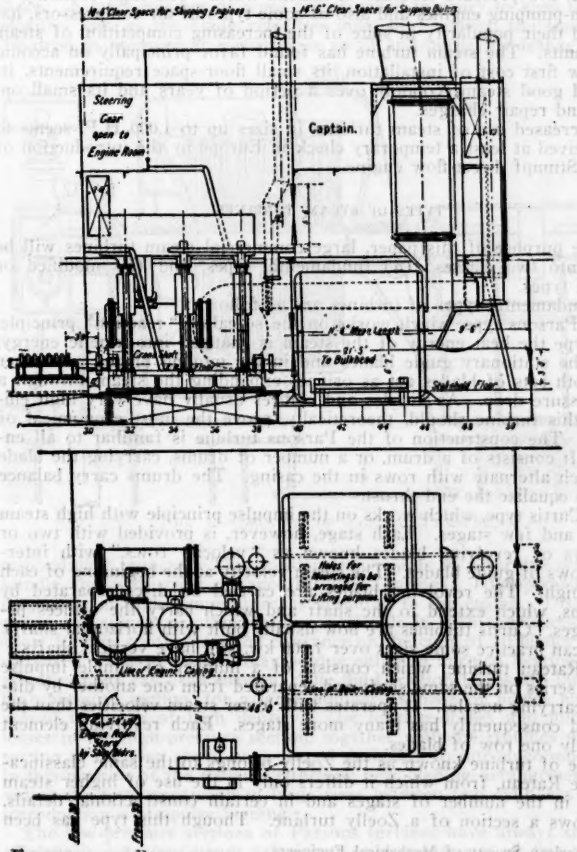


FIG. 8—1,300-H.P. STEAM

fuel, some of the bulkhead measurements and the wages of the hands, are those actually given us by the respective builders, and they may therefore be taken to be accurate as any figures which are to be obtained.

The price of the fuel has had to be assumed, as this varies much from time to time and according to the locality; the prices named, however, are a sufficiently close approximation to the average for home and foreign ports to give the desired comparison, but they can, of course, be altered to suit individual cases, and we think that the resulting figures which we give will prove of great interest. To us they have come as somewhat of a surprise in some instances, as we did not set forth to try and prove a case, but to learn the facts.—"The Engineer."


Steering Gear space to Engine Room
Captain.
To Bulkhead
Bulkhead Floor
Holes for Mountings to be arranged for Lifting purposes

ENGINE AND BOILER.

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES.*

By A. G. Christie.

PRESENT FIELD OF LARGE STEAM TURBINES.

Steam turbines are now being used for driving alternating-current generators, turbo-compressors, turbo-blowers, pumps and marine propellers, and by means of gearing, to furnish power to direct-current generators, rolling mills and the propeller shafts of steamships. Reciprocating engines were formerly used for such purposes, but recently this class of engine has seldom been installed except for rolling-mill work, non-condensing service as on heating systems, rope and belt drives, hoists, and in certain combinations with low-pressure turbines in marine work. The high economy of the piston-pumping engines and also of some types of air compressors, has continued their popularity in spite of the increasing competition of steam turbine units. The steam turbine has found favor principally on account of its low first cost of installation, its small floor-space requirements, its continued good steam economy over a period of years and its small operating and repair charges.

The increased use of steam turbines in sizes up to 1,000 H.P. seems to have received at least a temporary check in Europe by the introduction of the new Stumpf direct-flow engine.

TYPES OF STEAM TURBINES.

For the purpose of this paper, large commercial steam turbines will be divided into two classes: (*a*) fundamental types, and (*b*) modified or combined types.

The fundamental types of turbines are as follows:

a The Parsons type, which works on the so-called "reaction" principle. In this type the heat energy of the steam is changed into kinetic energy, both in the stationary guide blades and in the moving blades. In other words, both sets of blades act as orifices expanding the steam through a small pressure drop. As nozzles and orifices usually have very high efficiencies, this turbine should, theoretically, prove the most economical of all types. The construction of the Parsons turbine is familiar to all engineers. It consists of a drum, or a number of drums, carrying the blade rows which alternate with rows in the casing. The drums carry balance pistons to equalize the end thrust.

b The Curtis type, which works on the impulse principle with high steam velocities and few stages. Each stage, however, is provided with two or more rows of revolving blades known as "velocity rows," with intermediate rows of guide blades. The steam velocity at the beginning of each stage is high. The revolving blades are carried on discs separated by diaphragms, which extend to the shaft and which carry the orifices between stages. Curtis turbines are now usually built with horizontal shafts. In American practice some sizes over 7,000 kw. still have vertical shafts.

c The Rateau turbine, which consists of a number of simple impulse wheels in series on the same shaft and separated from one another by diaphragms carrying nozzles. It operates with lower steam velocities than the Curtis and consequently has many more stages. Each revolving element carries only one row of blades.

The type of turbine known as the Zoelly belongs to the same classification as the Rateau, from which it differs only in the use of higher steam velocities, in the number of stages and in certain constructional details. Fig. 1 shows a section of a Zoelly turbine. Though this type has been

* The American Society of Mechanical Engineers.

widely adopted abroad, it has not been placed on the American market up to the present time, except in certain torpedo-boat destroyers built at Cramps' shipyard.

Each of these fundamental types is based on sound theoretical principles. In the process of manufacture and in operation, certain features have not proved entirely satisfactory, hence far-reaching modifications have been made in the design of some types of turbines. Some manufacturers have combined the characteristics of two or more types to overcome the inherent limitations of each fundamental type. A discussion of the unsatisfactory operating conditions of each type will show the reasons which led up to the changes in recent turbines and will also aid in distinguishing the novel features of new designs.

The first rows of spindle blades in a standard Parsons turbine are placed on a drum of small diameter in order to make the blades as long as possible and to minimize the proportional leakage losses past the ends of the blades. A large number of rows are provided in order to keep the steam velocities low, as the blade velocities must be low with the small drum. The drop in pressure at each row is small and hence the leakage is correspondingly reduced. This construction results in a turbine with a long spindle and with great distance between bearings. High-pressure steam, frequently at high temperature, is admitted to the casing. Distortion of casing and spindle are thus easily conceivable in such construction, and to allow for this contingency the clearance on the ends of all blades is usually increased. This distortion may be due either to unequal heating or to the growth of the cast-iron casings. The fluid friction losses are large in this high-pressure section, for a large number of rows of blades must be revolved in steam of high density. The leakage losses and fluid friction losses in the high-pressure section, together with the troubles due to distortion in the long shaft, have forced designers to introduce modifications in this portion of the turbine, either by new blading arrangements, by dividing the total expansion in large sizes between two cylinders or by the introduction of impulse blading.

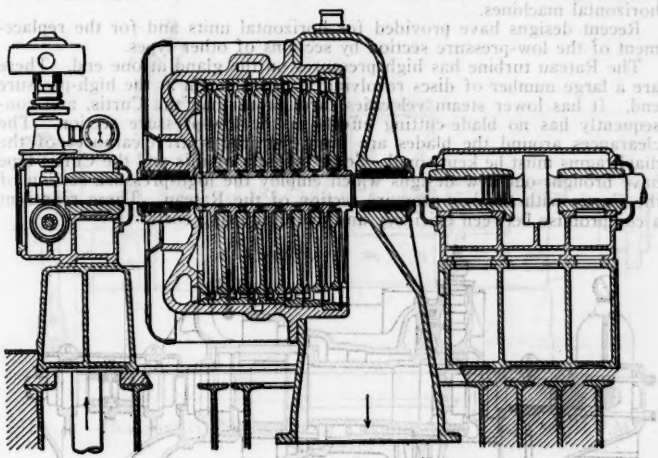


FIG. 1.—SECTION OF ZOELLY TURBINE.

The low-pressure sections of Parsons turbines have always shown high efficiencies. As low steam velocities are characteristic of this type, there

is no cutting away of blade material, even with very wet steam, provided no injurious properties are present in the feedwater. This low-pressure section has therefore been altered only in details.

The presence of a large low-pressure balance piston in close proximity to the steam inlet has frequently been the cause of serious distortions. Many builders now place this balance piston in the exhaust end, a construction due to Fullager.

The Curtis turbine utilizes high steam velocities in all stages. As steam becomes wet through expansion in the low-pressure stages, there is frequently considerable cutting of the blade material by the steam, although, contrary to first expectations, there is seldom cutting in the high-pressure blades due to the high initial velocities. The first row of velocity blades in a stage usually does the greater portion of the work, and hence the second row does not work at maximum efficiency. As the steam of the first stage is expanded very fully in the nozzles, there is no high pressure or superheat in the turbine casing or at the glands. The vertical type of unit is somtimes subject to electrical unbalancing and to other troubles peculiar to this construction. It is not as accessible in operation as the horizontal machines.

Recent designs have provided for horizontal units and for the replacement of the low-pressure section by sections of other types.

The Rateau turbine has high pressures on the gland at one end. There are a large number of discs revolving in dense steam at the high-pressure end. It has lower steam velocities throughout than the Curtis, and consequently has no blade-cutting effects in the low-pressure section. The clearances around the blades are large, but the shaft clearances of the diaphragms must be kept small. Some builders of this and the Curtis type have brought out new designs which employ the high-pressure section of the Curtis with the low-pressure section of the Rateau. These represent a compromise between efficiency and manufacturing costs.

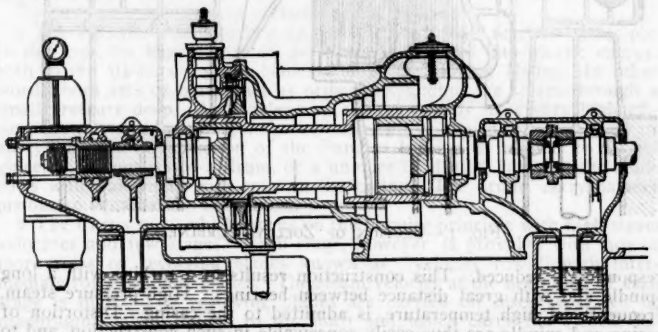


FIG. 2.—SECTION OF BROWN BOVERI'S CURTIS-PARSONS TURBINE.

Under the classification of modified or combined types, there are turbines with modified Parsons, Curtis and Rateau construction, and turbines with combinations of Curtis-Parsons, Rateau-Parsons, Curtis-Rateau and Curtis-Zoelly construction, with a few special combinations. Some typical turbines of these classes are described in the following paragraphs.

The Allis-Chalmers Company manufacture a turbine of the modified Parsons type. High peripheral speeds are employed with a decreased number of rows. A portion of the theoretical efficiency in the high-pressure end has been sacrificed by the use of fewer blade rows. Also a smaller proportion of work is done in this section than is usual in Parsons

turbines. The blades are all provided with a channel-shaped shroud. European experimenters have pointed out that better efficiencies are obtained with shrouded than with plain blades, as the so-called spilling over at the ends is prevented. Wing blades are fitted in the last low-pressure rows to take care of the large volumes of low-pressure steam. The spindle is much shorter and stiffer than the standard Parsons, and hence smaller clearances can be provided. The Fullager low-pressure balance piston is also used. The outstanding features of this design are, reaction principle with drum construction, few rows of blades with high steam velocities, short spindle construction, and employment of wing blades for high vacuum. The steam consumptions obtained on this type show improved efficiency over standard construction.

A number of Parsons turbines have been built in which the total expansion is divided between two cylinders. One of the best known units of this type is installed at Dunstan Power Station England, This was built by Richardsons Westgarth & Company on Brown-Boveri designs. This unit, however, employs a modified double-flow Parsons construction in the low-pressure cylinder.

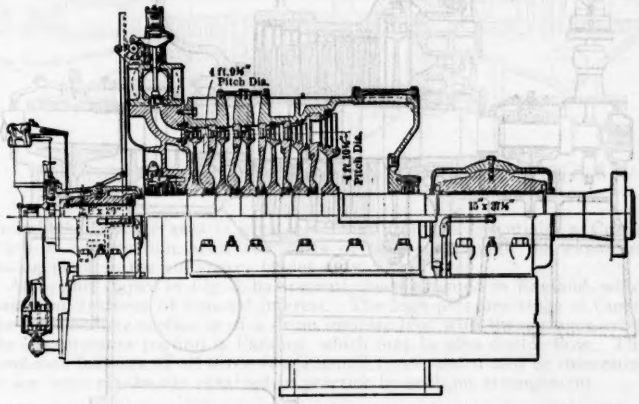


FIG. 3.—HORIZONTAL CURTIS STEAM TURBINE, 7,000 KW., 1,800 R.P.M., SIX-STAGE.

C. A. Parsons & Co. have used cast-steel cylinders for the high-pressure portion of their turbines so as to overcome the troubles due to deformation and growth of metal under high pressures and superheat experienced with cast-iron casings.

The Westinghouse Machine Company have developed a double-flow machine which employs a Curtis high-pressure stage with Parsons intermediate and double-flow low-pressure sections. There is only one balance piston in this machine next to the Curtis ring.

These turbines run at high speeds, have short shafts and small clearances. The steam is expanded in the nozzles, and hence there are no high pressures or temperatures in the casing. By providing two low-pressure sections, high vacuum can be economically utilized.

European builders of Parsons turbines, among whom are Brown Boveri & Company, C. A. Parsons & Co., Franco Tosi, Sulzers, Willans & Robinson and Erste Brünner, have replaced the high-pressure sections of their Parsons turbines by a Curtis wheel with two or more velocity rows but have retained the single-flow Parsons drum construction for the remaining por-

tions. One of these units is shown in Fig. 2. The temperatures and pressures in the casing are low as the steam is expanded in the nozzles. The distance between bearings is decreased, the shaft is stiffer and clearances are smaller than in the standard Parsons turbine. The Fullager balance piston is used in the turbines of several of these builders. Turbines built in this manner have shown some exceptionally good efficiencies.

Compared with the Westinghouse machine, the leakage at the end of the low-pressure blading is less than in the double-flow section. On the other hand, the Westinghouse machine has smaller balance piston losses and can also utilize the highest vacuum at better efficiency.

Melms & Pfenniger employ a drum impulse section of about five stages instead of a Curtis ring on some of their large turbines. They claim improved efficiency from this construction, though high-pressure and high-temperature steam are introduced into the casing.

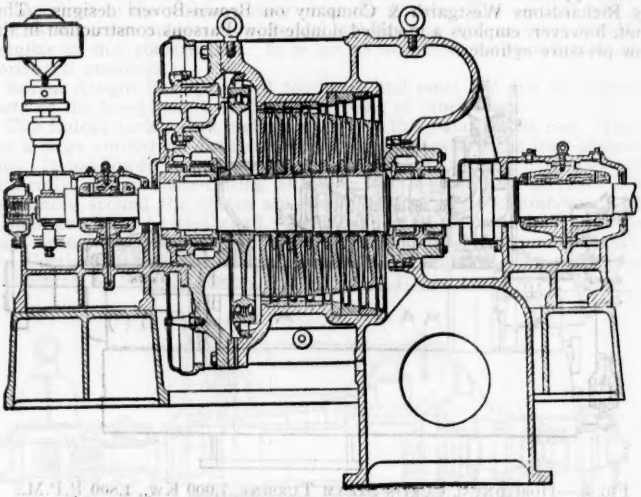


FIG. 4.—SECTION OF BERGMANN CURTIS-RATEAU TURBINE.

The General Electric Company now manufacture a horizontal type Curtis turbine in all sizes up to 7,000 kw., a section of which is shown in Fig. 3. This embodies all the essential features of the Curtis design. Compared with the vertical type, this design provides easier access to all working parts such as governor, bearings, valves, etc., and allows a better survey of the unit. The machine can also be dismantled and its internal parts examined with less trouble. The oiling problem is very simple compared with the vertical units, as there is no step bearing to provide for. It is possible that only horizontal units of all sizes will be built in the near future. The A. E. G. also builds similar turbines up to 1,000 kw. capacity.

Many manufacturers in Europe are now building a turbine of the type shown in Fig. 4. This consists of a Curtis high-pressure with Rateau or Zoelly low-pressure sections, the number of stages depending on the size of the machine, the speed and the steam conditions. The pressure and temperature in the casing are low, as a consequence of the expansion of the steam in the nozzles. The steam velocity is moderate in the low-pressure section. This machine is longer than the simple Curtis but shorter

than the Rateau. Its builders claim improved efficiencies over either of the fundamental types.

Belliss & Morcom, of Birmingham, England, have introduced a turbine which consists of a high-pressure Curtis stage with a low-pressure drum construction like Parsons, but fitted with impulse blading on the spindle and blades forming expansion parallel wall nozzles in the casing. All other impulse turbines employ diaphragms extending to the shaft. There are a greater number of rows in a Belliss turbine than in a corresponding Rateau turbine, and hence a lesser pressure drop through each set of guide blades. The net leakage loss may thus be lower than the Rateau, even

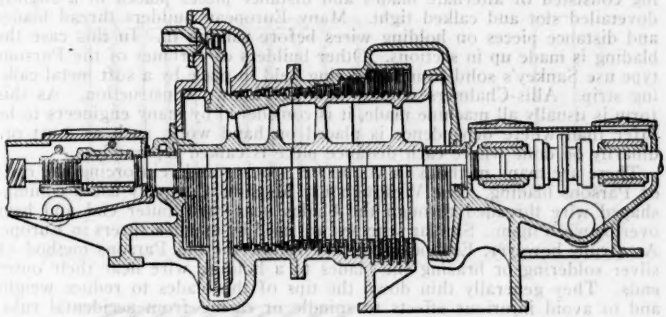


FIG. 5.—SECTION OF CURTIS DRUM-IMPULSE PARSONS TURBINE.

though the leakage area is greater. This turbine is essentially a Curtis-Parsons construction, with few rows of blades and with the expansion taking place in the stationary blades only.

A turbine, shown in Fig. 5, has recently been patented in England, which has some features of unusual interest. The high-pressure stage is Curtis, the intermediate section is of a drum impulse type with three stages, while the low-pressure portion is Parsons, which may be also double-flow. This combines features of all three fundamental types, and it will be interesting to see what results are obtained in practice by such an arrangement.

DISCUSSION OF TURBINE DETAILS.

In the early days of steam turbine building it was difficult to secure suitable materials to withstand the stresses set up at high speeds of rotation. But as the demand for such materials increased, much study was given to the requirements for this service so that it has been possible through the use of more suitable material to increase very considerably the speeds of all sizes of steam turbines.

Higher steam velocities are possible with increased peripheral speed, and thus fewer rows of blades or stages are required. This results in a shorter and more rigid shaft construction, which is therefore less liable to vibration. Many builders, especially those of Parsons turbines, have found this construction to give an increase in steam economy over the slow-speed types, so that the more compact modern high-speed machine is more desirable than the older type.

BLADING.

The requirements of a satisfactory blading material are that it shall withstand without deformation stresses due to centrifugal force, and temperature or pressure changes, it shall not cut out with high velocities of

steam, and that it shall withstand the corrosive effect of moisture. Parsons turbines have used special bronzes and copper-nickel alloys. Steel blades have also been used in some cases. In these turbines the steam velocities are low and there is usually no cutting on this account. The principal problm with this blading is to manufacture it cheaply and secure it in such a manner that it will withstand all stresses to which it is subjected. This blading design is therefore a question of detail, and, as shown in Table 1, many ingenious schemes have been devised.

In Parsons turbines the blades are usually cut, punched or pressed into proper form from strips of drawn material. The original Parsons blading consisted of alternate blades and distance pieces placed in a slightly dovetailed slot and calked tight. Many European builders thread blades and distance pieces on holding wires before calking in. In this case the blading is made up in sections. Other builders of turbines of the Parsons type use Sankey's solid foundation ring held in place by a soft metal calking strip. Allis-Chalmers employ this well-known construction. As this form is usually all machine made, it is considered by many engineers to be safer than where dependence is placed on hand work, such as must ordinarily be done where each distance piece is calked separately.

There are many methods in vogue for spacing and reinforcing the ends of Parsons blading. The Westinghouse Machine Company use a comma-shaped wire threaded through the blades near their outer end and bent over between them. Similar schemes are used by manufacturers in Europe. As a rule, however, European builders follow the old Parsons method of silver soldering or brazing the blades to a holding wire near their outer ends. They generally thin down the tips of the blades to reduce weight and to avoid injurious effects to spindle or casing from accidental rubs. Some builders, such as Sulzers, do not thin off their blades or use shrouds, but make their spindles so rigid and well balanced that the blades can be made with a very heavy cross-section and hence need no support. Several manufacturers rivet the outer ends of their blades into channel-shaped shroud rings. This gives an especially stiff construction.

Advocates of the shrouded blading claim that it provides a labyrinth passage for the steam and thus reduces the leakage losses from row to row. It also holds the blades at the required angles. It has been noted, however, that with wet steam there is a tendency for the moisture to pit the casing opposite the edges of the shrouds and thus increase the clearances. This action has also been noticed with unshrouded Parsons blading. The shrouded blading is usually so stiff that serious damage is done if rubbing starts between the blading and the spindle or casing.

The blading of impulse turbines is of nickel-steel, frequently with 25 per cent. nickel, in the high-pressure section and special bronze in the other stages. Experience with this 25 per cent. nickel-steel blading material has not been entirely satisfactory, and several manufacturers are now using a low carbon steel alloy with just sufficient nickel to prevent corrosion, usually about 5 per cent. These blades are said to be stronger and less liable to fatigue of material. Special bronze and monel metal have also been successfully used. These impulse blades are stamped from sheets, drop forged or milled from solid bars with or without a wide base to act as a distance piece, or are made from extruded metal strips of the desired cross-section. Usually these blades are of crescent section, but some are formed of flat strip material and made of constant thickness over the width of the blade. The separate distance pieces are usually of the same material as the blades themselves. In general, all impulse blades are provided with shrouds to prevent vibration and also to provide an enclosed passageway for the steam at high velocities. As there is no drop in pressure between the two sides of a row of moving blades, the clearance can be made large, both on the end and sides, so that there is little possibility of rubbing when in operation.

Impulse blading is usually held in place in dovetailed grooves or in tee-

shaped slots, although some manufacturers form their blades with two legs which straddle the discs and are held firmly in place by rivets.

The first impulse turbines had blading in which the inlet and discharge angles were equal. Now almost all builders use blading on which the discharge side of the blade makes a sharper angle with the axis than the inlet side. This does not necessarily mean that the discharge area of the blades is smaller than the inlet area, for the blade is usually lengthened radially on the discharge side. Thus both inlet and discharge areas are made equal. The sharper angle of discharge reduces slightly the relative velocity of exhaust from the moving blade. On turbines of the Curtis type there is

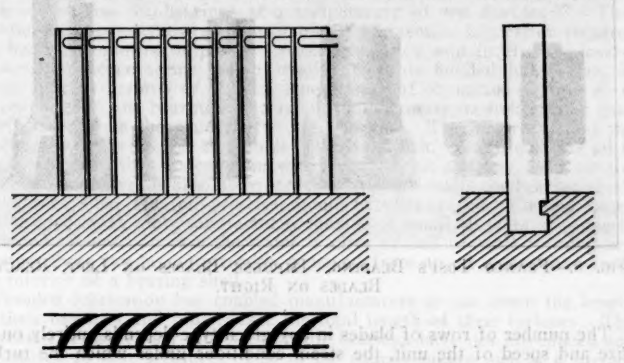


FIG. 6.—BROWN BOVERI'S REACTION BLADING.

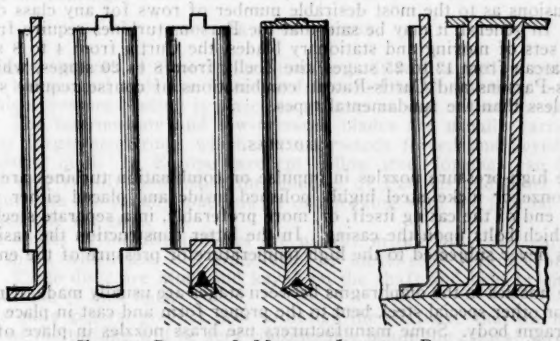


FIG. 7.—BELLISS & MORCOM IMPULSE BLADING.

usually only a small difference between the inlet and outlet angles of the first row of moving blades, but on the second row of moving blades in the stage, the entrance and exit angles often differ by as much as 15 degrees.

Some impulse turbine designs are such that there must evidently be some such reaction effect in the moving blades as is obtained in Parsons turbines, though not of sufficient amount to cause any noticeable end thrust.

Some recent interesting developments in blading are shown in Figs. 6 to 8. Brown Boveri's form of Parsons blading is illustrated in Fig. 6. Bel-

liss & Morcom use a unique form of blading on the drum impulse section. This blading is shown in Fig. 7. Franco Tosi cuts projections like threads in his blade grooves and mills corresponding projections on the base of the blades, as in Fig. 8. This is a very satisfactory but expensive form of blading.

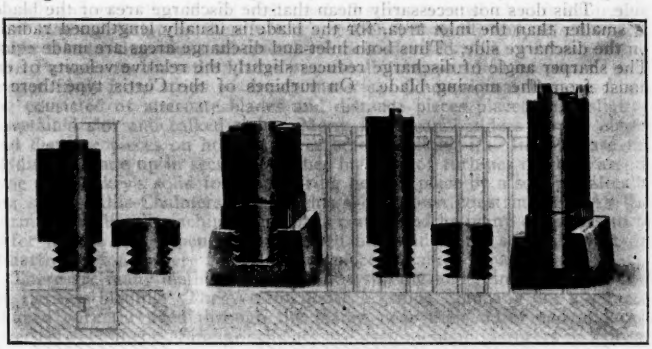


FIG. 8.—FRANCO TOSI'S BLADING. IMPULSE BLADES ON LEFT, REACTION BLADES ON RIGHT.

The number of rows of blades in any given type depends entirely on the size and speed of the unit, the steam conditions under which the turbine will operate and the heat drop per row or stage assumed in the design. Each designer selects such conditions as his experience and judgment lead him to consider the most satisfactory. It is thus hard to draw any general conclusions as to the most desirable number of rows for any class of turbine. In general, it may be said that the Parsons turbines require from 40 to 80 sets of moving and stationary blades, the Curtis from 4 to 8 stages, the Rateau from 12 to 25 stages, the Zoelly from 8 to 20 stages, while the Curtis-Parsons and Curtis-Rateau combinations of course require several rows less than the fundamental types.

NOZZLES.

The high-pressure nozzles in impulse or combination turbines are made of bronze or nickel-steel highly polished inside and placed either in the upper end of the casing itself, or, more preferably, in a separate steel casting which bolts upon the casing. In the latter construction the casing itself is never subjected to the high temperature or pressure of the entering steam.

The nozzles in the diaphragms between stages are usually made of nickel-steel or other special steel, bent to the proper form and cast in place in the diaphragm body. Some manufacturers use brass nozzles in place of steel made up in sections and riveted or bolted in place. All these passages must be smooth and preferably very highly polished to reduce friction losses. After the first stage all nozzles have parallel walls on the discharge side.

BEARINGS AND LUBRICATION.

Practice varies widely with regard to the design and construction of journals and bearings. Builders of impulse turbines invariably use cast-iron bearing shells provided with spherical self-aligning pads and lined with white metal. These are being used to an increasing extent on Par-

sons turbines and without exception on all low-speed units. Some manufacturers still retain the original Parsons form of bronze bearing shell with concentric rings on the outside, separated from each other by oil films. It has been claimed for this type of bearing that the oil films dampen any slight variation of the spindle and thus provide a quieter running machine. Experience has shown that this is not always the case and that such a bearing is often a real source of danger when the added clearance of the spindle due to play between the rings is taken into consideration. This construction is also much more expensive than the white metal bearings.

In Europe many turbines are run with a minimum amount of oil and with oil leaving the bearings at a temperature of 190 degrees F. This practice is based on the argument that such a system of lubrication requires the least expenditure of power for oil circulation and in friction losses. However, practice seems to be tending towards flooded lubrication, in which a great quantity of oil at a temperature of about 100 degrees F. is forced through the bearings by a pump of the rotary, centrifugal or gear type, driven from the main shaft of the turbine. The oil pressure at the bearings varies from 3 to 20 pounds per square inch. The life of the oil is much longer in this system than with very hot oil and any wear on the bearings themselves is absolutely prevented. Occasionally the bearing shells are water-cooled, but this practice should be discouraged. Cooling water can be used much more satisfactorily outside of small oil pipes in properly designed coolers. In case the water is dirty or full of scale-forming impurities, these can be more readily removed from an oil cooler than from the interior of a bearing shell.

Flooded lubrication has enabled manufacturers to cut down the length of their bearings and thus reduce the total length of their turbines. The increased pressures per unit area on the bearings have not introduced any difficulties, so that pressures of 80 to 100 pounds square inch at a surface speed of 60 feet per minute are common practice. The best results are usually obtained with a temperature of about 125 degrees F. as the oil leaves the bearings.

SPINDLE CONSTRUCTION.

Parsons turbines in America are usually built with a hollow quill into which the journal ends are forced and fastened by shrink links or bolts. The high-pressure blading is placed in grooves in one end of the quill itself. The intermediate and low-pressure blades are usually carried on cast or forged-steel rings which are afterwards forced and keyed upon the central quill. In Europe excellent hollow steel forgings can be obtained very readily, and hence the spindles of Parsons and other drum-type turbines are usually made up of one forging with the journal shafts fastened into the ends.

The shafts of impulse turbines are usually in one piece and carry the blade discs, which are high-grade steel or nickel-steel forgings or castings. These discs are fitted and keyed on the shaft and held in place by shrink links or lock nuts.

It is quite general practice now to design the shafts so that the normal speed of the turbine will be very considerably below the critical speed due to any slight unbalancing of the mass that may be present. This removes the dangerous vibrations often experienced when passing through critical speeds and permits closer clearances to be used on Parsons blading and in the labyrinth passages in the diaphragms between the stages of impulse turbines.

PACKING GLANDS.

A small impeller supplied with water is provided on all American-built Parsons turbines to form an air seal at the shaft glands. European builders prefer to use labyrinth packings with live or throttled steam as an

air seal. The objection raised by foreign builders to the water packing is that it takes too much power to drive the impeller, and that it provides a condensing surface for the steam. The amount of water required in a well-designed gland is very small, and there is no great circulation. Thus the water can have only a small effect as a condensing medium. Usually the steam directly inside the casing is under vacuum and then condensation would not be objectionable. It requires considerable steam to pack the labyrinth type of gland, and this loss often exceeds that due to the power required to drive the water impellers.

Impulse turbines use carbon rings at the high-pressure gland, especially when superheated steam is used. The leakage past the first rings is carried through a passage and pipe to the low-pressure glands to act as a vacuum seal. The labyrinth packing in the diaphragms is usually of bronze in the high-pressure stages and frequently of white metal in the low-pressure section. The low-pressure shaft glands are made either with carbon or labyrinth packing, sealed by steam.

THRUST BEARINGS.

Thrust bearings are now provided on all types of turbines. These serve to adjust the position of the spindle and to take up any end thrust present. The end thrust in well designed turbines is usually of small amount and seldom causes trouble. These bearings are usually flooded with oil and are often made with bronze rings which can easily be replaced in case of damage.

Several builders of Curtis-Parsons turbines have substituted an oil thrust piston for the steam balance piston of the usual form. This piston is placed on the governor end of the shaft outside of the casing proper. The thrust is taken up by oil which is supplied under pressure and which can escape only through sets of labyrinth baffles on the piston and on the surrounding chamber. The piston adjusts itself for the amount of thrust present by a small axial movement of the shaft, thus opening or closing the discharge area for the oil through the labyrinth baffles, and automatically varies the oil pressure on the face of the piston. This arrangement does away with the loss of steam through balance pistons and should improve the turbine efficiency, though a small amount of power is required to pump the oil.

GOVERNING DEVICES.

The speed of Parsons turbines is usually controlled by a centrifugal governor of the Hartung or similar type which regulates the position of a balanced poppet valve through the medium of a steam or oil relay. The latter type is coming into more general use on account of its many advantages over the steam relay. The steam is throttled either at constant pressure or by a pulsating action.

Impulse turbines and turbines employing a Curtis high-pressure ring usually govern by means of Hartung type governors and oil relays. The speed is sometimes controlled by simple throttling of the steam, which practice is common in Europe. In this case, additional nozzles can be opened or closed by hand as required. This system would be unsuitable with violently fluctuating loads. Other types employ both throttling and automatic nozzle regulation, while, again, many turbines, particularly of the Curtis type, are built for nozzle governing alone. In Europe there is a difference of opinion as to the most economical method of governing. M. H. Zoelly claims that he gets the best results on his type of turbine by throttle governing. It is generally admitted, however, that impulse turbines give the best results by controlling the number of nozzles that are open at any load. With such control the pressure before the nozzles in service is always the normal pressure for which these are designed. The General Electric Company govern their turbines above 300 kw. by means

of an oil-relay system which operates through a piston and camshaft to open or close nozzles as required by load conditions.

The Westinghouse Machine Company use a vibrating oil-relay system on many of their machines. Recently they have adopted on some sizes a very powerful governor which is direct-connected to the governing valve and operates it without the use of relays. This system was adopted by some of the early European builders, but was abandoned in favor of the oil-relay system.

Sulzers have recently installed turbines in which the conventional centrifugal force governor has been replaced by a hydraulic governor. This governor consists of a simple centrifugal pump, geared to the main shaft and delivering oil under pressure into a chamber beneath a spring-loaded piston. The pressure under this piston is thus dependent on the speed of the main turbine and the piston's position in its cylinder will vary accordingly. The piston is connected to the usual balanced valve of an oil relay which controls the oil supply from the main oil pump to the throttle valve in the usual manner. There are no mechanical parts to wear in this arrangement. The apparatus is extremely simple and has many distinct advantages.

All turbines are now provided with a small overspeed governor, usually placed at the outside end of the shaft. This operates at a determined percentage over speed and closes the main or secondary steam valve either by means of a steam or oil relay or by a falling weight through a system of levers and springs. The oil-relay system has the advantage that the valve shuts immediately should the oil supply fail for any reason.

Parsons turbines are usually provided with a secondary over-load valve which automatically admits live steam to the second diameter of blades. Impulse turbines have additional sets of nozzles with valves which may be opened automatically, or by hand in case of overloads.

CASINGS.

Practice varies widely in regard to the construction of turbine casings. These are generally made of cast iron, though some European builders make the high-pressure front end of cast steel. Parsons turbines are built with the top and bottom halves single castings or made in sections. Some of the older designs of impulse turbines used solid diaphragm plates placed on the shaft between the discs. The clumsiness of handling and the difficulty of making repairs with this construction has forced most builders to make these in halves and to fasten them to the top and bottom portions of the casing.

Cylinder casings are now made of symmetrical design and without any deep metal webs or ribs as stiffeners on the outside. Equalizing pipes with provision for expansion take the place of passages formerly cast in the casings themselves. Strains due to unequal temperatures must be avoided in all portions.

BED PLATES.

Several European builders make a practice of filling the hollow portions of their turbine and generator bed plates with concrete after erection. This adds more mass to the turbine unit and is said to dampen any slight vibration that may be present.

TURBO-GENERATORS.

A discussion of the design of turbo-generators is beyond the limits of this paper. In general, European builders of electrical machinery allow less overload capacity than is usual on American machines, but give better guarantees of efficiency and regulation.

Practice varies widely in regard to the normal rating and maximum

TABLE I DATA ON STEAM TURBINE CONSTRUCTION

Manufacturer	Type of Turbine	R.P.M. of Various Sizes	No. of Stages.	Blade Speed in Ft. per Sec.	Steam Velocities, Ft. per Sec.	Blade Material
C. A. Parsons & Co.	Parsons & Curtis-Parsons	.....	Usually about 70	270 on low-pressure blades	540	Special bronze and delta metal.
Westinghouse Mach Co.	Parsons & Curtis-Parsons	Up to 3000 kw., 3600; 3000-5000 kw., 1800; over 5000 kw. (750 or 720)		Curtis blades 600; Parsons blades 350	Curtis 2000; Parsons 600	Special bronze and soft steel; Curtis blades bronze
Willans & Robinson	Curtis-Parsons	.....	1 Curtis; 12-20 rows Parsons on 1 diameter of drum	.....	.....	Bronze and copper-nickel blades
Brown Boveri & Co., Richardsons Westgarth & Co.	Parsons & Curtis-Parsons	Up to 3000 kw., 3000; 3000-7500 kw., 1500	.....	Curtis blades 460, Parsons blades 380	.....	Curtis blades of special alloy; Parsons blades of bronze
Brush Electrical Engrg. Co.	Curtis-Parsons	.....	.....	.....	.....	Parsons blades bronze alloy
Erste Brünner M. F. G.	Curtis-Parsons	Up to 1500 kw., 3000; 1500-4000 kw., 1500; 4000 kw., up, 960	1 Curtis; remainder Parsons on 2 diameters of drums	.....	.....	Both Parsons and Curtis blades of special bronze
Franco Tosi.	Curtis-Parsons	Up to 2500 kw., 3000; 2500-8000 kw., 1500; 8000-12,000 kw., 1000	1 Curtis; remainder Parsons on 1 diameter of drum	.....	Curtis 2130; Parsons 230-460	Parsons and Curtis blades of special bronze
Gebrüder Sulzer.	Curtis-Parsons	.....	.....	Curtis blades 360; Parsons blades 310	.....	Curtis blades all of special steel; Parsons blades of special bronze
Melms & Pfenninger, Breitfeld, Danek & Co.	Rateau-Parsons & Curtis-Parsons	Up to 4000 kw., 3000; 4000-9000 kw., 1500	.....	Up to 575	Impulse section up to 2200; Parsons up to 900	Rateau blades of special bronze; Parsons blades also special bronze
Allis-Chalmers Co.	Parsons	Up to 2000 kw., 3600; 2000-5000 kw., 1800	.....	Up to 370	400-800	Copper-nickel alloy
Allegemeine Elektricitäts Gesellschaft.	Curtis-Rateau	Up to 3000 kw., 3000; 3000-9000 kw., 1500; 9000 kw. and over, 1000	1 Curtis; 9-12 Rateau	.....	.....	Bronze alloy or nickel-steel

Maschinenfabrik Augsburg Nurnberg.	Curtis-Zoelly	Up to 6000 kw., 3000; 6000-10,000 kw., 1500; 10,000-15,000 kw., 1000	1 Curtis; 6-8 Zoelly	Up to 650	Curtis 2500; Zoelly about 1300	Nickel-steel highly polished
British Westinghouse Co.	Curtis-Rateau	Up to 1500 kw., 3000; 1500-5000 kw., 1500	1 Curtis; 6-12 Rateau	300-500	Curtis 1500; Rateau 800	5% nickel-steel
Bergmann Electricitäts Werke.	Curtis-Rateau	.....	1 Curtis; 8-12 Zoelly	410-460	Curtis 2600; Rateau 1170	25% nickel-steel
Bellise & Morcom.	Curtis-Drum Impulse	Up to 1500 kw., 1800; use lower speeds than most builders	1 Curtis; 25-30 rows of impulse blades on drum	Not exceeding 400	.....	Curtis blades of steel; Impulse blades of phosphor bronze
Escher, Wyss & Co.	Zoelly	Up to 1500 kw., 3000; 1500-3500 kw., 1500; over 3500 kw., 1000	3000 r.p.m., 10 stages; 1500, 16 stages; 1000, 20 stages	.....	About 1400	5% nickel-steel
Maschinenfabrik Oerlikon.	Rateau	Up to 4000 kw., 3000; 4000-7500 kw., 1500; over 7500 kw., 750	10-16 stages with 2 or more mean diameters of blades	320-520	1310-1700	Nickel-steel
Sachsische Maschinenfabrik.	Zoelly	.....	5-6 on first diameter, 6-8 on second diameter	.....	.....	Nickel-steel
Frazer & Chalmers.	Rateau	.....	Are reducing number of stages on all turbines	Have increased blade speeds recently	.....	Nickel-steel
Pokorny & Wittekind.	Curtis-Rateau & Rateau	.....	.....	500-600	2300 admission	.....
General Electric Co.	Curtis	Up to 1400 kw., 3600; 1400-7000 kw., 1800; 7000-20,000 kw. (750 or 720)	4-7	460-500	About 1200 average	Special bronze extruded metal; Monel metal and steel used on some low-pressure blades
British Thomson Houston Co.	Curtis	Up to 1000 kw., 3600; 1000-5000 kw., 1800; up to 2000 kw., 3000; 2000-7500 kw., 1500	Same construction as General Electric Co.	600 as maximum	About 1800 from nozzles, 1100 from diaphragms	Special bronze and steel

TABLE 1—*Continued*

Manufacturer	Blade Form	Method of Blade Manufacture and Insertion
C. A. Parsons & Co.	Parsons form with very thick rounded inlet, blades thinned off at tips	Alternate distance pieces held by calking, ends of blades stiffened by silver soldering to a brass strip; Curtis blades shrouded
Westinghouse Mach. Co.	Blades crescent with heavy central portion and sharp inlet edge, dovetail base on impulse	Alternate distance pieces held by calking, ends stiffened by comma wire threaded through blades and bent over
Willans & Robinson.	Parsons rounded inlet, stamped to fit Sankey foundation ring and channel shroud; usual Curtis blades on impulse	Parsons sections made up by machine and held in place by calking in soft metal strip, final riveting of shroud completed last
Brown Boveri & Co., Richardsons Westgarth & Co.	Parsons blades not thinned off, usual form, base of blade notched to fit corresponding lateral projection in the grooves, blades not calked	Curtis blades in dovetail slot, no sharp corners on base and shrouded; Parsons blades threaded on wire and silver soldered
Brush Electrical Engrg. Co.	Usual Parsons form with calked distance pieces in dovetailed groove; also use grooved base of blades fitting corresponding grooves in shaft like teeth in a bolt	Tips of blades thinned off, blades silver soldered to wire strip
Erste Brünner M. F. G.	Curtis blades have special dovetail base and are separate from their distance pieces; Parsons blades usual form with distance pieces in dovetail groove	Curtis blades shrouded; Parsons blades thinned at tips and silver soldered to wire on side
Franco Tosi.	Blades and distance pieces milled at base, with thread-like projections to fit corresponding grooves in spindle	Curtis blades shrouded; Parsons blades also have channel shroud-riveted over after blades are in place
Gebrüder Sulzer.	Parsons blades of very heavy cross-section with standard Parsons distance piece and method of calking	Parsons blades not thinned at ends or reinforced in any way; Curtis blades milled to fit dovetail slot
Melms & Pfenninger, Breitfeld, Danek & Co.	Impulse blades have slot in side to fit projection in groove and held in place by soft metal calking strip; Parsons blades have special base to fit foundation ring and with special shroud on end	Parsons blade rings made up in halves and fitted to machine, sections held in place by soft metal calking strip, angle of blades, 20 to 45 deg.

Allis-Chalmers Co.	Parsons section with more defined inlet edge, blading uses Sankey foundation ring and shroud, blading made in sections outside turbine	Sections held in dovetailed slots by soft metal calking strips, shrouds channel-shaped
Allegemeine Elec tricitäts Gesell schaft.	Blade crescent shaped but of lighter section toward exhaust end, shrouded in groups	Blades cut from solid bar and set in dovetailed grooves with separate distance pieces
Maschinen fabrik Augsburg Nurnberg.	First Curtis rows have blades of heavy section, remaining rows are thin in section, blades are held in a ⊥ slot	Blades mostly drop-forged, all rows shrouded, outlet and inlet angles vary considerably
British Westinghouse Co.	Curtis blades have ⊥ base to fit corresponding groove; flateau blades have 2 projections at base straddling disk and rivets rolled in place	Blade and distance pieces milled from solid steel bars
Bergmann Electricitäts Werke.	Blades stamped from sheet metal with projections at base straddling disk and riveted to it, blades of uniform thickness across section	Distance pieces drop-forged of special bronze and riveted with blades to disk, all blades shrouded
Belliss & Morcom.	Curtis blades, usual form; on impulse section blades are of uniform thickness over section and shrouded	Blades made with strip at base bent in form of L, distance piece formed with opening at base, the 2 projections spread over base of blade and fill dovetailed slot when calked down
Escher, Wyss & Co.	Blades stamped from sheet metal with ⊥ base, blades sometimes thinned off toward tips, distance piece stands well above surface of shaft	Blades milled with sharp edges and highly polished after stamping, distance pieces milled from solid bar, only thin shroud ring used
Maschinenfabrik Oerlikon.	Blades punched to shape from drawn metal strips, base is ⊥ shaped, simple band shroud ring used	Distance pieces milled, slot closed by special locking piece
Sachsische Maschinenfabrik.	Zoelly blades thinned off toward tips on low-pressure end, base ⊥ shaped with rounded corners	Blades and distance pieces milled from solid bar, channel-shaped shroud
Frazer & Chalmers.	Blades with forked base riveted to disks	Blades milled from solid bar, no shroud ring, but head milled on blades like De Laval
Pokorny & Wittekind.	High-pressure blades shrouded, low-pressure blades unshrouded on some units	.....
General Electric Co.	Blades milled with dovetail base and have shroud ring, distance pieces also milled	Blades put in place in special blading machine
British Thomson Houston Co.	Same construction as General Electric Co.	Same construction as General Electric Co.

TABLE 1—*Continued*

Manufacturer	Nozzles and Diaphragms	Shaft and Spindle Construction	Form and Material of Bearings
C. A. Parsons & Co.	.....	Spindles usually a single hollow forging, shaft ends fastened in	Parsons concentric bronze sleeve bearings in self-aligning cast-iron casings, large sizes white metal lined
Westinghouse Mach. Co.	.....	Shaft, hollow quill with blade rings forced on, ends held in by special patented fastening	Parsons concentric sleeve bearing on small units, cast-iron self-aligning bearings, white metal lined on large units
Williams & Robinson	.....	Disk, solid forging; drum, hollow rolled forging; ends fastened in by locked bolts; Fullager balance piston used	Cast-iron, white metal lined, self-aligning
Brown Boveri & Co., Richardsons Westgarth & Co.	Nozzles made in 3 parts of steel or bronze and all machined	Disk, solid forging forced and fastened on hollow forged drum spindle; shaft ends locked into drum; Fullager balance piston used	Parsons concentric bearings on high speeds, cast-iron, white metal lined, self-adjusting bearings on low speeds
Brush Electrical Engrg. Co.	.....	Disk, solid and fastened on solid or hollow forged drum spindle	Cast-iron, white metal lined, self-adjusting bearings on all sizes
Erste Brünner M. F. G.	Nozzles of special bronze alloy	Small sizes, nickel-steel, solid rotors; large sizes, hollow forged high grade steel; Curtis ring bolted to spindle	Parsons concentric sleeve bearings on small units; on large units cast-iron white metal lined self-aligning bearings
Franco Tosi.	Nozzles of steel or bronze, angles of nozzles 20 to 25 deg.	Forged steel shaft with spindle rings and impulse disk with balance piston forced on and held by bolts	Cast-iron, white metal lined, self-aligning bearings
Gebrüder Sulzer.	Nozzles of sheet brass cast in valve chest	Spindle, hollow forging with shaft ends and impulse disk bolted on	Cast-iron, white metal lined, self-aligning bearings
Melms & Pfenninger, Breitfeld, Danek & Co.	Nickel-steel guide blades in Rateau section with special labyrinth packing, angle of nozzles 18 deg. to 22 deg.	Spindle, hollow forging of nickel-steel, impulse disk and balance piston fastened on; shaft ends forced in	Parsons concentric sleeve bearings on small units, large units have cast-iron, white metal lined, self-aligning bearings
Allis-Chalmers Co.	.....	Hollow forged quill with blade rings of cast steel forced on; shaft ends forced into quill and held by bolts and buts; Fullager balance piston	Cast-iron, white metal lined, self-adjusting bearings

Allegemeine Electricitäts Gesellschaft.	Nozzles of bronze or nickel-steel, guide blades in diaphragms of nickel-steel cast in place	Solid shaft with disks forced on; disks of steel tapering toward the blades	Cast-iron, white metal lined bearings, usually water-cooled; only 3 bearings; very small governor bearing carried by front housing
Maschinenfabrik Augsburg, Nurnberg.	Nozzles of nickel-steel milled from solid, guide blades in diaphragms of highly polished sheet nickel-steel	Short stiff solid shaft running below critical speed disks of soft steel forced on shaft and locked; disks taper toward blade ends	Cast-iron, white metal lined, self-adjusting bearings; only 3 bearings used; governor bearing very small
British Westinghouse Co.	Nozzles of special steel, guide blades of special steel cast in place in diaphragms	Short, solid, stiff shaft stepped from center to receive disks which are of forged steel tapering from center to periphery	Cast-iron, white metal lined, self-adjusting bearings
Bergmann Electricitäts Werke.	Nickel-steel nozzles in 4 groups, nickel-steel guide blades cast in place in diaphragms	Short, solid shafts not too stiff, as shafts at 3000 r.p.m. and over, run above critical speed; disks forced on shaft and locked.	Cast-iron, white metal lined, self-adjusting water-cooled bearings
Belliss & Morcom.	Nozzles, steel or bronze, impulse guide blades of brass made in segments	Forged steel drum fastened to through solid shaft; drum carries impure disk and balance piston; heavy rigid construction.	Cast-iron, white metal lined, self-adjusting bearings
Escher, Wyss & Co.	All nozzles and guide blades of nickel-steel cast in place and highly polished	Solid shaft with disks supported on ⊥ ring so as to make shaft less rigid	Cast-iron, white metal lined, self-adjusting bearings
Maschinenfabrik Oerlikon.	Nickel-steel guide blades all cast in place	Nickel-steel shafts made heavy in the center; disks of forged soft steel forced on and locked to shaft	Cast-iron, white metal lined, self-aligning bearings, water-cooled in the larger sizes
Sachsische Maschinenfabrik.	Sheet-steel guide blades cast in place	Solid shaft with nickel-steel disks, highly polished, forced on	Cast-iron, white metal lined, self-adjusting, water-cooled bearings
Fraser & Chalmers.	Diaphragm guide blades cast in place	Solid shaft with disks forced on	Cast-iron, white metal lined, self-adjusting bearings
Pokorny & Wittekind.	Angle of nozzles 20 deg.	Solid shaft with disks forced on	Cast-iron, white metal lined bearings
General Electric Co.	Inlet nozzles of special bronze, steel guide blades in diaphragms cast in place and polished	Shaft, solid forgings; disks, cast steel forced on shaft	Cast-iron, white metal lined, self-adjusting bearings with copper cooling pipes cast in babbitt metal lining
British Thomson Houston Co.	Same construction as General Electric Co.	Shaft and disks of steel	.....

TABLE 1—*Continued*

Manufacturer	Surface Speed of Journals, Ft. per Sec.	Bearing Pressure, Lb. per Sq. In.	Type of Oil Pump	Governor and Method of Governing
C. A. Parsons & Co.	35-50	40-50	.....	Steam relay system, steam admitted in puffs, not continuous throttling
Westinghouse Mach. Co.	40-50	50-70	Rotary oil pump with sliding gate	Direct throttling from governor, sometimes pulsating; also pulsating oil relay
Willans & Robinson.	45-60	50-70	Both reciprocating and rotary types used	Direct throttling from governor
Brown Boveri & Co., Richardsons Westgarth & Co.	.....	.....	Rotary pump	Hartung type with oil relay, also slight pulsation to valve; valves on nozzles open automatically
Brush Electrical Engrg. Co.	.....	.....	Rotary pump	Steam relay system, pulsating action
Erste Brünner M. F. G.	50	40-50	Double rotary valveless pump	Hartung type governor, steam relay and piston valve on main inlet; pulsating governing
Franco Tosi.	.....	55-90	Rotary gear pump	Hartung type governor, oil relay system on main valve, automatic oil relay nozzle governing
Gebrüder Sulzer.	.....	.....	Triple gear pump	Automatic hydraulic regulator, centrifugal pump on oil only; throttling of supply; oil relay on all gear; no mechanical governor
Melms & Pfenninger, Breitfeld, Danek & Co.	40-80	35-85	Eccentric rotary pump	Hartung type oil relay, throttling governor
Allis-Chalmers Co.	50-60	60-140	Rotary lobe pump, auxiliary duplex pump for starting	Hartung type oil relay, throttling governor
Allegemeine Electricitäts Gesellschaft.	50-70	100-125	Rotary gear	Hartung type oil relay, governor throttling steam; also automatic nozzle control

Maschinenfabrik Augsburg, Nurnberg.	.....	45-70	Horizontal rotary gear	Hartung type oil relay, governor throttling steam; also automatic steam control on nozzle valves
British Westinghouse Co.	.....	.....	Rotary gear pump	Heavy Hartung type governor direct-connected to throttling valve
Bergmann Electricitäts Werke.	.....	.....	Rotary gear pump	Hartung type governor throttling steam: nozzle valves opened by hand
Belliss & Morcom.	40-50	As high as 90	Double-acting valveless pump, 80-100 r.p.m.	Centrifugal governor, oil relay simply throttling steam supply
Escher, Wyss & Co.	.....	.....	Gear pumps, 1 for relay, 1 for bearing oils	Hartung type, oil relay simple throttling governor
Maschinenfabrik Oerlikon.	Not exceeding 66	Not exceeding 75	Rotary gear pump	Hartung type, oil relay governor, throttling steam supply; main throttle valve also opened by oil relay
Sachsische Maschinenfabrik.	.....	.....	Rotary type	Oil relay governor
Frazer & Chalmers.	.....	.....	.....	.....
Pokorny & Wittekind.	50-80	30-50	.....	Oil relay governor
General Electric Co.	.....	.....	Rotary gear pumps	Vertical centrifugal governor operating oil relay which controls nozzles; nozzle governing
British Thomson Houston Co.	50-60	50-70	Rotary pump	Same as General Electric Co.

TABLE 1—*Continued*

Manufacturer	Oil Data	Type of Glands	Provision for Overspeed Regulation
C. A. Parsons & Co.	.....	Steam-packed labyrinth at shaft	.....
Westinghouse Mach. Co.	Oil pressure on bearings 3 to 4 lb., relay pressure 60 lb.	Water-packed glands with centrifugal impeller	Emergency governor to shut off steam at about 12% overspeed
Willans & Robinson.	Bearing pressure about 10 lb.	Steam-packed labyrinth	Emergency governor for overspeed
Brown Boveri & Co., Richardsons Westgarth & Co.	.....	Steam-packed labyrinth	Emergency governor closes spring controlled throttle valve
Brush Electrical Engrg. Co.	.....	Steam-packed labyrinth; water-packed glands also used	Emergency oil relay control on throttle valve
Erste Brünner M. F. G.	7-10 lb. on bearing	Steam-packed labyrinth	Emergency governor to close spring loaded throttle valve
Franco Tosi.	10.4 gal. per min. pumped on 300 kw.; 18.4 gal. per min. on 12,000 kw.	Steam-packed labyrinth; water-packed glands also used	Emergency oil relay governor to close main throttle valve at 15% overspeed unique device
Gebrüder Sulzer.	Bearing oil pressure 20 lb.	Steam-packed labyrinth; labyrinth packing consists of large number thin sheets placed radial, with close clearance on shaft	Oil relay control on steam inlet valve
Melms & Pfenninger, Breitfeld, Danek & Co.	Oil temperature at bearing 105 to 120° F.	Steam-packed labyrinth; exhaust end packing adjustable.	Emergency valve to operate at from 10-15 per cent overspeed
Allis-Chalmers Co.	Bearing pressure 5 lb., relay pressure 30 lb., bearing oil temperature 110° F. best; large quantities pumped	Water-packed glands with centrifugal impeller	Emergency governor to close spring-loaded throttle valve

Allegemeine Electricitäts Gesellschaft.	Oil temperatures at bearings from 175 to 190° F.	Steam-packed labyrinth and carbon rings	Emergency governor to close spring-loaded throttle valve
Maschinenfabrik Augsburg, Nurnberg.	.....	Graphite rings in glands, steam-packed	Emergency governor
British Westinghouse Co.	.....	High-pressure end, labyrinth and water gland; low-pressure end, water gland with centrifugal impeller	Emergency governor closes spring actuated safety valve on steam inlet
Bergmann Electricitäts Werke.	Relay pressure 30 lb.	Steam-packed labyrinth	Emergency governor to close throttle at 10% overspeed
Bellias & Morcom.	Relay pressure 40 lb.; estimate 10 per cent of heat losses to go through oil, design oil pump accordingly	Multi-cellular with carbon rings and steam packed	Emergency governor controls oil relay to shut off steam
Escher, Wyss & Co.	.....	High-pressure end, carbon ring packing; low-pressure, labyrinth	Emergency governor to operate at 8-10% overspeed
Maschinenfabrik Oerlikon.	Oil usually required is 0.02 gal. per kw-min.	High-pressure, graphite; low-pressure, steam-packed labyrinth	Emergency governor closes main throttle valve through oil relay
Sachsische Maschinenfabrik.	.....	High-pressure, carbon; low-pressure, steam-packed labyrinth	Emergency governor to operate at 15% overspeed
Frazer & Chalmers.	.....	High-pressure, metal rings and labyrinth; low-pressure, labyrinth	.....
Pokorny & Wittekind.	From 8-18 gal. per min. circulated according to size	High-pressure, carbon rings and labyrinth; low-pressure, labyrinth	.....
General Electric Co.	Relay pressure about 70 lb., bearing pressure 25 lb.	Carbon ring packing on both ends	Emergency governor to close unbalanced throttle valve
British Thomson Houston Co.	⅛ gal per min. per bearing on 1000 kw. units; 1¼ gal. on 5000 kw. unit; same amounts to thrust	Carbon ring packing	Emergency governor to close throttle valve at 15% overspeed

TABLE 1—*Continued*

Manufacturer	Type of Coupling	Additional Notes
C. A. Parsons & Co.	Flexible claw	Simple Parsons turbines, usually very long spindles; large sizes in 2 cylinders
Westinghouse Mach. Co.	Flexible claw	Small sizes simple Parsons; large sizes Curtis-Parsons, double flow in low-pressure, heavy thrust block to take any end thrust
Willans & Robinson.	Flexible claw	.....
Brown Boveri & Co., Richardsons Westgarth & Co.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum, 5-6% per 1 in. vacuum; (*b*) steam pressure, 1% per 15 lb.: (*c*) superheat, 1% per 11° F.
Brush Electrical Engrg. Co.	Flexible claw	.....
Erste Brünner M. F. G.	Flexible claw	
Franco Tosi.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum, 7% per 1 in. at 28 in.; (*b*) steam pressure, 1% per 15 lb.; (*c*) superheat, 1% per 12.5° F.; large thrust block in form revolving piston, labyrinth packings; oil admitted both sides, movement shaft opens one side, closes other, making pressures on both different, automatically compensating for end thrust
Gebrüder Sulzer.	Special design of flexible coupling	End thrust all taken up by automatic oil controlled balance piston on end of shaft similar to Franco Tosi's
Melms & Pfenninger, Breitfeld, Danek & Co.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum 28-29 in., 5%; 27-28 in., 4%; 26-27 in., 3.5%; (*b*) steam pressure 1% per 12 lb.; (*c*) superheat 1% per 10° F.
A'lis-Chalmers Co.	Flexible claw	.....

Allegemeine Electricitäts Gesellschaft.	Solid	.....
Maschinenfabrik Augsburg, Nurnberg.	Solid	.....
British Westinghouse Co.	Flexible claw	.....
Bergmann Electricitäts Werke.	Solid	.....
Belliss & Morcom.	Special design of flexible coupling	Flood bearings with great quantity of oil to flush away heat rapidly
Escher, Wyss & Co.	Solid	.....
Maschinenfabrik Oerlikon.	Flexible with leather disks	Blade angles, 20 deg. on stationary blades, 30 deg. on moving blades
Sachsische Maschinenfabrik.	Flexible with leather disks	.....
Fraser & Chalmers.	.....	.....
Pokorny & Wittekind.	Solid	Practically all turbines built for connection to turbo-compressors; effect on steam consumption of changes in (a) vacuum live-steam turbines, 5% per in., exhaust steam turbines 10% per in. vacuum; (b) steam pressure, 1% per 25 lb.; (c) superheat 1% per 12° F.
General Electric Co.	Solid	.....
British Thomson Houston Co.	Solid	Effect on steam consumption of changes in (a) vacuum 7% increase per in. above 28 in., 150 lb. steam gage, 8% decrease per in. below 28 in., 150 lb. steam gage; (b) steam pressure 1% per 10 lb. around 150 lb. pressure; (c) superheat 1% per 10° F.

capacities of different turbines. Some European builders guarantee their generators to carry normal full load without undue heating only for two to six hours. Others follow the practice which has been introduced in America of rating their turbines at the maximum load they will carry continuously. Several builders still offer turbines which will carry as high as 25 per cent. overload continuously.

COMMERCIAL CONSIDERATIONS.

Each turbine is designed to operate at maximum efficiency at some given vacuum. Owing to uncertainties in design factors for losses and to slight inaccuracies in construction, the most efficient vacuum may vary somewhat from that for which the turbine was designed. Theoretically, the steam consumption should decrease as the vacuum increases, but this rate of decrease will vary for each type of turbine, depending on the blade areas and steam velocities of the low-pressure section. It is, therefore, essential that this rate of change be determined for each individual turbine by actual test.

In Table 1 some data are presented in tabulated form regarding various types of steam turbines. This information was gathered during personal visits to the various works and also by correspondence. Designs of steam turbine details are constantly changing, so that many of the items in this table may not correctly represent the latest practice of the various builders.

RESULT OF TESTS.

If the steam could expand freely to exhaust pressure in a turbine without radiation, friction, eddy or windage losses, its expansion would be adiabatic and on the Rankine cycle. The "efficiency ratio" expresses the proportion of the heat actually turned into work to that available from such an adiabatic expansion. In other words, it expresses the efficiency of the actual turbine as compared to the ideal turbine, and is independent of the type of turbine.

The B.T.U. per kw.-hour is figured above the heat of the liquid at exhaust pressure. This is not a satisfactory standard by which to compare results, for it is largely dependent on conditions beyond the control of the turbine builder. For instance, if the plant does not contain superheaters, the B.T.U. per kw.-hour will be high. The same will be true of a plant which has a warm cooling-water supply for condensers and consequently carries low vacuum. Yet the turbines may be designed to give a high efficiency ratio under these conditions. In fact, they may be able to utilize the heat available more efficiently than the turbines in another plant with both high superheat and high vacuum.

This can be seen in Table 2, in which recent turbine tests have been tabulated. The Brown-Boveri turbine at the Dunstan power plant uses 14,980 B.T.U. per kw-hour with an efficiency ratio of 68.8 per cent. Yet the Westinghouse City Electric with a lower steam pressure, lower superheat and lower vacuum, has an efficiency ratio of 68.9 per cent., though using 16,925 B.T.U. per kw-hour. The Erste Brünner Vienna turbine requires 16,460 B.T.U. per kw-hour with 71.8 per cent. efficiency ratio. It is therefore apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine.

The test results in Table 2 were grouped in order to analyze the relative merits of the different types of turbines on the basis of efficiency ratios. The Curtis-Parsons machines built by Erste Brünner hold first place in the list, but are followed closely by others of the same type built by Brown Boveri and Westinghouse Machine Company. The Parsons turbines built by Allis-Chalmers and Brown Boveri also show high efficiencies. The second class in the order of efficiency includes turbines of the Curtis-

Rateau and Curtis-Zoelly types, among which the turbines of the A. E. G. and British Westinghouse Company show remarkably good results. The next group includes simple Zoelly and Rateau turbines. The last group comprises straight Curtis types.

The superiority of the Curtis-Parsons over the Parsons type is probably due to the reduction in the fluid friction and rotational losses occurring in the first cylinder of the Parsons by the use of a Curtis stage in this section.

The Parsons low-pressure sections evidently utilize the heat in the steam only slightly more efficiently than do the impulse turbines. The great surface areas of all disc-type turbines which must be whirled in steam, produces losses which are apparently somewhat larger than the combined whirling losses and leakage in the Parsons drum turbines. Both the Zoelly and Curtis-Rateau types appear to use the steam more effectively in the low-pressure sections than the Curtis alone. Many European engineers hold the opinion that where high economy is to be obtained, the impulse turbine of the Rateau or Zoelly type is superior to the Curtis, though its manufacturing costs are higher. The Curtis-Rateau construction has all the commendable features of impulse turbines and has proved very economical.

The results shown in Table 2 are from the best reliable tests that have been made on each type. Objection may be raised that these results do not represent actual operating conditions as under varying loads, nor the average economy of any type of turbine. For instance, the Curtis turbine usually gives a very flat water-rate curve, while the Parsons type is more convex. On the other hand, recent tests on the new Curtis-Parsons types have also shown flat water-rate curves at various loads. It was impossible to compare the various types from this standpoint on account of absence of complete data of such tests.

It is interesting to note in Table 2 that the best results have been obtained within the past two years and that these show a considerable increase in efficiency over the earlier turbines.

LOW AND MIXED-PRESSURE TURBINES.

Many low-pressure turbines have been erected since 1904 and have shown very economical results. In some installations it is usual to provide for operation on high-pressure steam when the supply of exhaust steam is insufficient to meet the power demand. Hence, the mixed-pressure turbine has been developed. In Europe a Curtis stage is added at the inlet and the live steam passed through this before entering the low-pressure section. The whole of the heat content of the live steam can be effectively utilized by this method. Bleeder turbines are also being built, in which, after partial expansion to some fixed pressure, a portion of the steam is withdrawn from the casing for heating or industrial purposes.

Low-pressure turbines are frequently installed to use the exhaust steam of reciprocating engines without regenerators. In this case the generators are sometimes tied together electrically and the turbines are only fitted with an overspeed governor. Such turbines are usually installed in stationary work, only when the reciprocating engines are already in service. The high-pressure turbine in a new plant requires less floor space, has less complicated machinery, is cheaper in first cost and in maintenance, and approaches, if it does not equal, the economy to be derived from the combination unit in every-day service. Nevertheless, there have been a number of combined engine and turbine plants recently installed in England which have proved very satisfactory.

It is probable that low-pressure turbines will be installed in the near future in large gas-engine stations to utilize the waste heat in the gas-engine exhausts.

TABLE 2 ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES

Efficiency Ratios based on E.H.P. Marks & Davis Steam Tables used

Maker of Turbine	Type	Date of Test	Load-Kw.	R.p.m.	Steam Pressure Lb. Absolute	Temperature at Throttle, deg. fahr.	Vacuum referred to 29.92" Bar	Condenser Pressure, Lb. Absolute	Lbs. of Steam per Kw-Hr.	B.t.u. per Kw-Hr.	Heat Utilised per Lb. of Steam	Heat Available per Lb. of Steam	Efficiency Ratio	Reference
Erste Brünner M. F. G.	Curtis-Parsons	1910	2128	1500	156.2	482	27.89	0.995	13.82	16460	247.0	343.8	71.8	Periodische Mitteilungen
Erste Brünner M. F. G.	Curtis-Parsons		6000	960	184.9	573	28.18	0.854	12.56	15570	271.5	380.7	71.3	Zeit. D.V.D. Ing., 12/10/'10
Erste Brünner M. F. G.	Curtis-Parsons	1910	7442	960	192.0	584	28.18	0.853	12.625	15705	270.2	384.4	70.3	Periodische Mitteilungen
Westinghouse Machine Co.	Curtis-Parsons	1910	9173	1800	181.7	433	27.81	1.032	14.57	16925	234.1	340.2	68.9	Trans. A.S.M.E., vol. 32
Brown Boveri & Cie.	Curtis-Parsons		3053	1360	150.2	505	29.00	0.456	13.01	15990	262.2	385.5	68.0	Dinglers P.J., 6/17/'11
Erste Brünner M. F. G.	Curtis-Parsons	1910	1416	1260	128.2	482	27.60	1.137	15.18	18060	224.6	326.5	68.8	Periodische Mitteilungen
Brown Boveri & Cie.	Curtis-Parsons	1911	1750	1500	176.4	586	27.08	1.392	14.23	17500	239.5	354.8	67.5	Zeit. F.D.G. Turb., 5/30/'11
Brown Boveri & Cie.	Curtis-Parsons	1910	3764	1500	161.2	561	28.77	0.562	13.04	16290	261.5	391.4	66.8	Zeit. F.D.G. Turb., 5/30/'11
Westinghouse Machine Co.	Curtis-Parsons		9830	750	192.2	475	27.22	1.322	15.15	17790	225.2	336.0	67.0	Trans. A.S.M.E., vol. 32
Brown Boveri & Cie.	Curtis-Parsons	1911	1495	3000	200.6	563	26.41	1.720	14.78	17880	230.7	345.5	66.8	Data from Manufacturer
Brown Boveri & Cie.	Curtis-Parsons	1911	1271	3000	172.1	568	27.31	1.278	14.61	17880	233.6	354.3	65.9	Data from Manufacturer
Westinghouse Machine Co.	Curtis-Parsons		11466	750	191.7	484	28.07	0.910	14.45	17210	236.0	360.5	65.5	Trans. A.S.M.E., vol. 32
Erste Brünner M. F. G.	Curtis-Parsons		1250	3000	184.9	573	27.89	0.996	14.32	17680	238.2	373.1	63.9	Zeit. D.V.D. Ing., 12/10/'10
Brown Boveri & Cie.	Curtis-Parsons	1910	3320	1500	180.9	525	29.02	0.440	13.50	16680	252.7	401.3	63.0	Zeit. F.D.G. Turb., 5/30/'11
Brown Boveri & Cie.	Curtis-Parsons		5128	1000	171.2	565	28.52	0.726	14.35	17830	237.7	382.9	62.1	Stodola, 4th ed., p. 449
Breitfield, Danek & Co.	Impulse-Parsons	1909	3585	896	160.7	457	28.32	0.782	16.08	19070	212.0	352.4	60.2	Zeit. D.V.D. Ing., 12/10/'10
Brown, Boveri & Cie.	Parsons	1910	6257	1210	203.7	559	29.02	0.440	11.95	14980	285.5	415.0	68.8	Official Test Report
Allis-Chalmers	Parsons	1908	4300	1800	186.4	484	27.96	0.960	14.02	16690	243.4	355.7	68.4	Sibley Jour. of Eng., 1/11'
Brown Boveri & Cie.	Parsons	1903	3500	1360	156.4	499	28.84	0.532	13.71	16720	248.5	378.6	65.6	Zeit. D.V.D. Ing., 12/10/'10
Brown Boveri & Cie.	Parsons		3000	1360	165.0	625	27.02	1.120	14.75	18433	231.3	359.5	64.3	Die Turbine, 6/20/'11
C. A. Parsons & Co.	Parsons		5164	1200	214.3	509	28.95	0.473	13.18	16140	258.7	402.3	64.3	Stodola, 4th ed., p. 439
Allis-Chalmers	Parsons	1911	3850	1800	164.7	491	27.91	0.983	15.40	18410	221.3	348.3	63.5	Power, 1/2/'12
A. E. G.	Curtis-Rateau	1911	6518	1220	198.7	601	29.28	0.352	11.43	14640	298.4	434.2	68.7	Official Test Report
A. E. G.	Curtis-Rateau	1911	6565	1220	200.2	597	29.18	0.406	11.64	14848	293.0	427.7	68.5	Official Test Report

British Westinghouse	Curtis-Rateau	1911	5066	1500	190.2	552	28.68	0.649	13.00	16100	262.4	391.5	67.0	Electrical Review, 6/23/'11
M. A. N.	Curtis-Zoelly		3584	1500	178.3	569	27.54	1.166	13.99	17190	243.7	361.3	67.5	Data from Manufacturer
Bergmann	Curtis-Rateau	1909	1545	1500	188.5	581	28.59	0.654	12.97	16230	263.0	396.3	66.4	Zeit. D.V.D. Ing., 12/10/'10
Bergmann	Curtis-Rateau	1910	2477	1500	140.0	522	28.81	0.588	13.93	17135	244.8	373.4	65.6	Elec. Zeit., 4/20/'11
A. E. G.	Curtis-Rateau	1908	4239	1500	188.3	662	29.11	0.397	11.97	15620	284.9	439.0	64.9	Stodola, 4th ed., p. 404
British Westinghouse	Curtis-Rateau	1911	2930	1500	210.2	568	28.18	0.894	13.72	16935	248.7	383.3	64.9	Electrical Review, 4/28/'11
A. E. G.	Curtis-Rateau	1907	3169	1500	184.7	592	29.11	0.397	12.74	16230	267.7	425.1	63.0	Trans. A.S.M.E., vol. 32
M. A. N.	Curtis-Zoelly		2507	1500	175.5	460	27.40	1.234	16.24	19020	210.0	334.6	62.8	Data from Manufacturer
Bergmann	Curtis-Rateau	1909	1562	1500	186.8	555	28.33	0.780	14.57	17970	234.1	381.3	61.4	Data from Manufacturer
James Howden & Son	Zoelly	1909	6383	1000	202.7	520	27.33	1.269	14.305	17150	238.5	353.0	67.5	Engineer, London, 10/29/'09
M. A. N.	Zoelly	1910	1400	3000	180.7	554	27.40	1.237	14.21	17310	240.0	356.2	67.4	Zeit. D.V.D. Ing., 12/10/'10
Escher, Wyss & Co.	Zoelly	1910	2052	3000	193.9	585	28.39	0.750	13.04	16290	261.5	392.6	66.6	Zeit. F.D.G. Turb., 2/20/'11
Escher, Wyss & Co.	Zoelly	1910	4189	1000	179.7	557	28.66	0.618	13.30	16520	256.5	391.3	65.5	Zeit. F.D.G. Turb., 2/20/'11
F. Ringhoffer	Zoelly	1908	3000	1000	170.7	470	27.60	1.138	15.52	18278	219.8	339.2	64.8	Zeit. D.V.D. Ing., 12/10/'10
M. A. N.	Zoelly	1910	1250	3000	182.1	582	28.82	0.540	13.09	16500	260.2	404.5	64.4	Zeit. D.V.D. Ing., 12/10/'10
Oerlikon	Rateau	1911	3166	1500	213.9	663	29.25	0.367	11.44	14970	298.2	450.6	66.1	Engineering, 10/20/'10
Escher, Wyss & Co.	Zoelly		5118	1000	133.7	549	27.55	1.161	15.18	18530	224.6	341.6	65.7	Dinglers P.J., 7/15/'11
Escher, Wyss & Co.	Zoelly	1908	5000	1000	166.4	539	26.38	1.736	16.13	19350	211.2	330.4	63.9	Zeit. D.V.D. Ing., 12/10/'10
Escher, Wyss & Co.	Zoelly		3540	1500	155.1	469	28.21	0.838	15.07	17940	226.3	349.5	64.8	Dinglers P.J., 7/15/'11
Escher, Wyss & Co.	Zoelly	1910	1641	3000	221.0	672	27.91	0.985	13.08	16775	260.6	406.5	64.1	Zeit. F.D.G. Turb., 2/20/'11
Escher, Wyss & Co.	Zoelly	1910	1235	3000	176.8	451	28.39	0.750	15.35	18156	222.3	357.8	62.2	Zeit. F.D.G. Turb., 2/20/'11
British Thomson-Houston	Curtis	1911	2987	1500	154.7	605	26.75	1.557	15.96	18960	213.7	321.2	66.5	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis		3464		210.0	513	28.75	0.575	13.62	16620	250.4	393.4	63.6	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1909	2500	1500	126.5	414	28.47	0.711	15.92	18590	214.0	336.1	63.7	Zeit. D.V.D. Ing., 12/10/'10
A. E. G.	Curtis	1906	3000	1500	191.3	590	29.05	0.427	12.79	16240	266.6	420.4	63.4	Zeit. D.V.D. Ing., 12/10/'10
A. E. G.	Curtis	1909	2236	1500	191.6	654	29.34	0.284	11.77	15450	289.8	455.8	63.6	Zeit. D.V.D. Ing., 12/10/'10
Gen. Elec. Co.	Curtis		8880		192.5	487	28.02	0.933	15.05	17965	226.7	359.5	63.1	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1911	1541	1500	149.7	365	27.97	0.956	17.46	19720	195.3	320.2	61.0	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis		10816	750	190.0	525	29.39	0.260	12.90	16135	264.5	427.3	61.9	Trans. A.S.M.E., vol. 32
Gen. Elec. Co.	Curtis		5095		185.1	554	29.40	0.255	12.71	16090	268.4	436.0	61.6	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1911	1221	3000	134.7	448	27.16	1.353	17.75	20690	192.2	314.0	61.2	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis	1910	8775	750	194.0	451	27.95	0.956	15.95	18720	213.8	350.8	61.0	Trans. A.S.M.E., vol. 32

References: Zeit. D.V.D. Ing.—Zeitschrift des Vereines Deutscher Ingenieure; Zeit. F.D.G. Turb.—Zeitschrift für das Gesamte Turbinenwesen; Dinglers P.J.—Dinglers Polytechnisches Journal; Elec. Zeit.—Electrotechnische Zeitschrift.

TURBO-COMPRESSORS.

Turbo-compressors have some decided advantages over reciprocating compressors, such as smaller floor space, absence of inlet and discharge valves, low cost of upkeep and no internal lubrication. They are being manufactured quite extensively in Europe and have been introduced in America by the General Electric Company.

Turbo-compressors are built either with curved or radial impeller blades which discharge the air into smooth expanding diffusor channels to convert the velocity energy of the current into pressure. Usually guides are provided to direct the air into the entrance of the next stage without eddies. All passages are made as smooth as possible with no abrupt bends or turns, and all walls are water-cooled. The air is prevented from leaking back from stage to stage by labyrinth packings such as are used in impulse turbines.

European builders have installed turbines to deliver air at as high pressures as 130 pounds per square inch gage and are prepared to furnish them up to 180 pounds discharge pressure.

Turbo-blowers have also been built to deliver large volumes of air at low pressure, such as are required in furnace work. These units have no water-cooled jackets. The first difficult problem encountered in the construction of turbo-blowers or compressors was the provision of a suitable governing device for the unit. However, several ingenious and satisfactory arrangements have recently been developed and it is probable that this difficulty will soon be overcome completely.

The efficiency of turbo-compressors with water-cooling is defined as the ratio of the power required to compress the given quantity of gas isothermally to the power consumed at the compressor coupling in the actual compression. This efficiency in well designed units with discharge pressures between 60 and 150 pounds should fall within the limits of 60 and 70 per cent. The best results noted up to the present time were obtained on a turbo-compressor built by Pokorny and Wittekind for the Victoria Falls Power Company in South Africa, which on official test showed an efficiency of 67.7 per cent.

When there is no water-cooling provided, the efficiency of a turbo-blower is expressed as the ratio of the power required to compress the given quantity of air adiabatically to the power actually expended at the compressor coupling. This efficiency, depending on the size of the blower, should fall between 70 and 80 per cent. as a maximum. An efficiency of 78 per cent. has been obtained on official tests of a Rateau turbo-blower built by Kuhnle, Kopp and Kausch and is probably the best result obtained up to the present time on this type of compressor.

It can thus be seen that in so far as efficiency is concerned, the turbo-compressor is equal to the average reciprocating compressor. It seems probable that turbo-compressors and blowers will be used to an increasing extent, largely on account of their low first cost and operating costs as compared with steam reciprocating units.

The high thermal efficiency of the gas-driven blowing engine exceeds that possible in a turbo-compressor unit, so that the former will continue to be used in blast-furnace work.

TURBO-DRIVEN PUMPS.

The steam turbine is an ideal source of power to drive centrifugal pumps, especially when it is necessary to lift against high heads. Hence it has been installed in several places for city fire service, using lake or river water in high-pressure mains. The efficiency of such centrifugal pumps usually ranges from 65 to 80 per cent., so that, in spite of the high efficiency of the turbine itself, the combined set will not give as good

economy as a high-grade reciprocating pumping engine. However, its first cost is low, it requires no internal lubrication, takes up very little floor space, and has no valves to require examination or renewals.

GEARED TURBINES.

Attempts have been made to adopt steam turbines for direct connection to continuous-current generators and other slow-speed machinery. A steam turbine to be economical must be a high-speed machine, and hence its use with slow-speed machinery has not proved entirely satisfactory. Dr. De Laval adopted spur gearing as a means of reducing speeds on his first simple impulse turbines, and builders of this type still use this construction. It is only within the last few years that attempts have been made to apply gearing as a means of speed reduction on other types of turbines. The Westinghouse Machine Company are now manufacturing direct-current turbo-generators with the Melville-MacAlpine reduction gearing between turbine and generator. This gear is also being built for use in marine work to drive slow-speed propellers. The gear wheels in this construction are carried on a floating frame so that the teeth may be always in correct alignment. One set of such gearing showed under test an efficiency of 98.5 per cent.

C. A. Parsons & Co. have built several notable reduction gears for steam turbines, in which the floating-frame idea was omitted. A mixed-pressure steam turbine of 750 B.H.P. is now in use driving a three-high set of rolls through gearing at the Calderbank Steel Works near Glasgow, Scotland. A flywheel is placed on the same shaft as the driven gear and thus takes the shock off the turbine when a billet enters the rolls. In 1909 the Parsons Marine Steam Turbine Company installed a geared turbine of 1,000 H.P. in the S. S. *Vespasian* of 4,000 tons. Extensive experiments were carried out and it was found that the efficiency of this gearing, which had no floating frame, ranged between 98 and 99 per cent. After a year's operation, in which the ship covered 20,000 miles, tests were again made on the gearing with equally good results. On examination no appreciable signs of wear could be noted on the gear teeth, which were made of mild chrome-nickel-steel and were flooded with oil.

It is perfectly feasible to adapt the steam turbine through gearing to belt and rope drives when these are required in large powers. Under certain conditions, where fuel costs are high and water for condensation is plentiful, such an installation would prove an economical investment in place of reciprocating engines. The direct turbine drive would then come into competition with the motor drive. The losses in line shafting can be greatly reduced by the use of ball or roller bearings so that the turbine drive may prove very economical in some instances as compared with individual motor drives.

The use of geared turbines in large sizes has been in the nature of an experiment until quite recently. Judging from the results obtained in recent installations, their commercial success seems now assured. In the design of these units the tendency will be to simplify details. The success of the Parsons gears on the S. S. *Vespasian* should encourage designers to do away with any special devices to secure alignment and to provide simply accurately cut gears properly meshed and running in a flood of oil.

MARINE TURBINES.

All the standard types of turbines have now been adopted for marine service in driving screw propellers, either direct-connected or through gearing. The design of direct-connected turbines is complicated by the fact that the speed of screw propellers must necessarily be low as compared with the most favorable speeds for economical steam-turbine operation. Hence these turbines require large spindle diameters and massive

construction and yield correspondingly poor steam economies, especially at slow speeds. These units are usually built with the power divided between two or more shafts connected to high and low-pressure cylinders. With geared equipments the turbines can be operated at their most efficient speeds, while the gears can be so designed that the propeller also runs at its most economical speed.

Reversing is made possible by suitable blading in the low-pressure ends of the main units into which live steam is admitted when desired. When the turbine is running forward, this blading revolves in vacuum and consumes but little power.

Combined types of turbines are also being introduced in marine installations. A recent Curtis design includes a drum impulse section. M. Zoelly now uses Curtis stages in his high-pressure section, but with steam velocities not exceeding 1,300 feet per second, obtained by converging nozzles only, and drum impulse construction on the low-pressure ends. Several other European builders have also used Curtis stages in the high-pressure portion.

Reciprocating engines exhausting into low-pressure turbines have been installed in several ships, the most notable of which is the *Olympic*, and have shown very satisfactory results. In this case the engines are used for reversing.

Many schemes have been proposed to install turbo-generators of central-station type on shipboard and to operate the propeller shafts by means of large slow-speed induction motors. Marine engineers object to this arrangement on account of the dangers accompanying the use of such electrical machinery and auxiliaries in marine service. This objection seems to be due largely to inexperience with electrical machinery, as the essential conditions of operation in marine work do not differ greatly from those under which many electrical machines operate satisfactorily in land practice.

Turbo-driven lighting sets and other auxiliaries are being used in increasing numbers on shipboard, owing largely to the high efficiencies which may be obtained, to the small floor space required, and to the light weight of the units.

Recent orders for turbine-driven steamships abroad include some interesting equipments. The Canadian Pacific Railway has ordered two boats with four screws and with Parsons turbines. The two outside screws will have high and intermediate-pressure turbines respectively, while the two center screws will be connected to low-pressure turbines. Two twin-screw passenger boats of 5,000 H.P. with Parsons turbines have been ordered for the Southampton-Havre service. Each set consists of a high-pressure and a low-pressure turbine geared individually to its own propeller shaft. The British Government has ordered two twin-screw destroyers also to use Parsons geared turbines, totalling 14,000 H.P. per ship, or 7,000 H.P. per gear. The United States Government has placed an order with the Westinghouse Machine Company for a gear equipment on one of its colliers.

Geared turbine equipments are thus making rapid headway on account of the high efficiency of the combination and the resultant favorable steam consumption obtained.

At the present time about 90 per cent. of the marine turbines built have been of the Parsons type. Here again the inefficiency of the Parsons high-pressure sections has become apparent, so that it is probable that a construction similar to Zoelly's high-pressure end will be introduced in this section. Sir Charles A. Parsons is quoted as saying: "In the low-pressure blades of the *Mauretania* the leakage was practically nothing and their efficiency was about 85 per cent." Under such conditions it does not seem probable that much higher efficiency can be obtained by use of other constructions than the Parsons reaction type in the low-pressure section. It is reported that the low-pressure blading of Parsons turbines

in ships of the United States Navy has given considerable trouble, and also that the turbines need more careful handling when starting up than do impulse turbines.

TREND OF TURBINE DEVELOPMENT.

The cost of manufacture is a very important item in determining the future development of the steam turbine. Types such as the original Parsons and the Rateau, while inherently of very high efficiency, have too high manufacturing costs to compete with the newer combined types.

The writer offers as his opinion that the combined types, such as the Curtis-Parsons, the Curtis-Rateau and also the Curtis-Rateau-Parsons, previously described, will very soon supersede the simple types. It is probable that the Curtis turbine will eventually be built only in horizontal units and will gradually be modified to a Rateau or even a drum impulse construction in the low-pressure sections. The freedom from close adjustment in impulse turbines and the recent improvements in blading materials will greatly increase the use of this type, although Curtis-Parsons turbines are said to be cheaper to manufacture. In actual operation, it is an open question among engineers whether the reaction turbine has a higher commercial efficiency than the impulse type, and hence buyers usually consider first cost and personal preference only.

Turbines will probably be made shorter with very stiff shafts. With this construction many of the earlier blading troubles will disappear. But the peripheral speeds will also be increased and this will involve the development of suitable blading material and methods of holding blades that will satisfy these new requirements. Recent results seem to indicate that improved efficiency may be looked for with increased blade speeds.

Several impulse turbines have been built recently in Europe where the expansion was not complete in the nozzle, so that a portion of the expansion took place in the first moving blades. Some large Curtis turbines recently installed in America are said to have Parsons blading in the last stage. These developments would indicate a movement to introduce reaction principles in impulse turbines, and further illustrate the tendency to merge types.

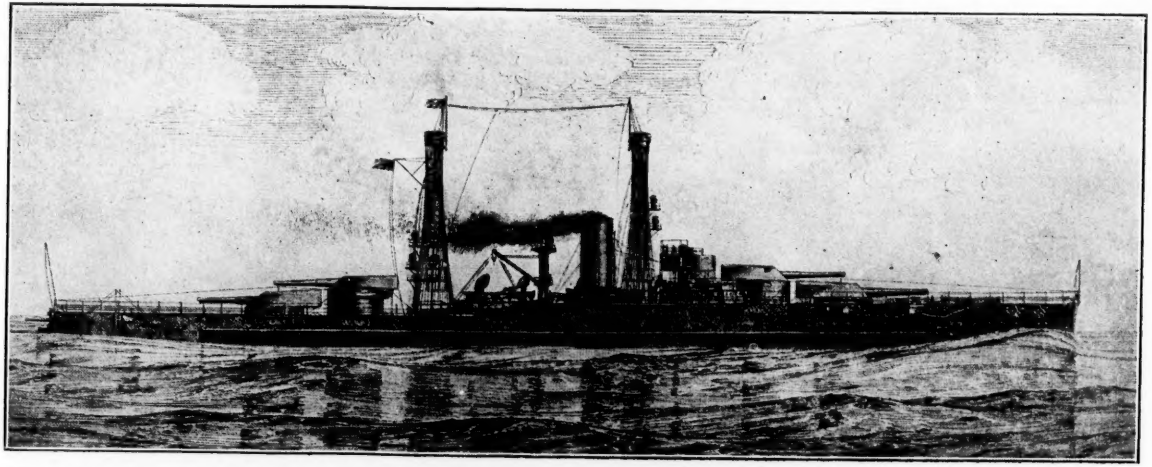
The hope of further improvement in efficiency lies in extensive study, particularly of the action of the steam during its passage through the moving and stationary blades, of the effect of form of blades, passages and casings and of various forms of baffles and balance pistons to prevent leakage. Such research work has not been carried out up to the present time by most manufacturers, largely on account of the extreme care and heavy expense involved in such tests. The present state of development has been largely one of cut and try. The increasing competition of the gas engine and the possible development of a satisfactory gas turbine will force manufacturers to develop their turbines to the greatest degree of economy.

With regard to detail, simplicity will be the leading consideration. With the introduction of Curtis high-pressure stages, nozzle governing will undoubtedly be used to an increasing extent, though the results obtained by Westinghouse, by Zoelly, and by Bergmann with simple throttling governors, raise a question as to whether the additional complication of nozzle governing will pay. Oil relays will probably replace all other systems of governing on account of their simplicity and reliability. The simple and efficient centrifugal oil-pump governor of Sulzer appears to be an improvement of considerable moment, and will probably receive extensive use.

With the development of suitable gearing for steam turbines, their field of application has been greatly increased and turbines will shortly be used for purposes which engineers today would consider them utterly

unfit. Low-pressure turbines will continue to be installed in plants where reciprocating engines are still in operation and also where large quantities of waste heat are available. The low and mixed-pressure types of turbines will find a very extended use in connection with heating systems, evaporators, etc.

The development of the past ten years has been truly marvellous. No great gain in thermal efficiency seems possible, so that future improvements will be largely along the line of detail construction and modification.



Length, 583 feet. Beam, 95 feet 2⅛ inches. Draught, 28 feet 6 inches. Displacement, 27,500 tons. Fuel, oil exclusively. Speed, 20¼ knots.
Armament, ten 14-inch guns. Torpedo tubes, four 21-inch. Armor belt, 13½ inches; barbettes, 13 inches; turrets, 9 to 18 inches.
Smokestack, 13 inches. Conning tower and tube, 16 inches. Gun deck, 3 inches. Protective deck, 2 inches.

THE NEW UNITED STATES BATTLESHIPS "NEVADA" AND "OKLAHOMA."

SHIPS.

UNITED STATES.

OUR LATEST BATTLESHIPS, THE *NEVADA* AND *OKLAHOMA*.

THE MOST POWERFULLY PROTECTED SHIPS YET DESIGNED.

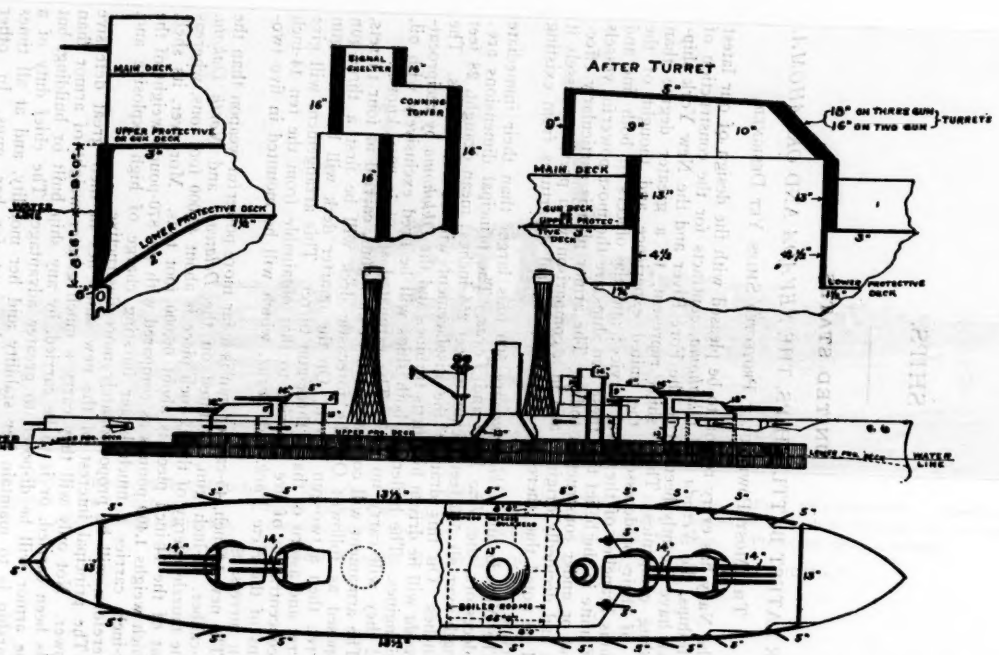
The Navy has every reason to be pleased with the design of our latest battleships, the *Nevada* and *Oklahoma*, contracts for the construction of which have recently been let to the Fore River and the New York shipbuilding companies. These ships represent, to a greater degree than any of their predecessors, the united experience and thought of the various branches of the naval service; and the officers of both line and staff unite in the belief that these two ships are the most powerful vessels afloat or under construction today. The armor plan is particularly effective and decidedly original, and in a comparison with previous vessels it will be noted that there are some very radical departures from existing practice.

The *Nevada* and *Oklahoma* are 500 tons larger than their immediate predecessors, the *New York* and *Texas*. The principal dimensions are: Length over all, 583 feet; beam, 95 feet 2⅝ inches; mean draught, 28 feet 6 inches. On this draught the displacement will be 27,500 tons. The *Nevada* will be driven by Curtis turbines and the *Oklahoma* by reciprocating engines. The boilers in both ships will be fired exclusively with oil, and they will carry no coal. The estimated speed is 20½ knots.

The armament will consist of ten 14-inch guns, carried in four turrets, disposed as follows: On the forecastle deck will be first a three-gun turret, then a two-gun turret. On the quarter deck will be a two-gun turret and astern of that a three-gun turret. This arrangement will give a concentration of fire superior to that obtainable from the ten 14-inch guns of the *New York* and *Texas*, which will be mounted in five two-gun turrets.

The new 14-inch, 45-caliber gun is a far more powerful weapon than the 45-caliber, 12-inch gun, mounted on the *Delaware* and *North Dakota*. The muzzle energy of the 12-inch piece is about 49,000 foot tons, whereas that of the 14-inch piece is about 66,000 foot tons. Moreover, its shell, which weighs 1,400 pounds as compared with the 870-pound weight of the 12-inch, carries a much larger bursting charge of high explosive and, therefore, will be proportionately more destructive.

The principal interest of the new ships lies in their great defensive power. Not only will they carry a much greater weight of armor than has been carried, or is to be carried, by any ship built or building, but the armor will be disposed to greater advantage. The chief duty of a warship is to maintain her stability and her mobility, and at all times present a completely-protected emplacement for her guns. In other words, she must not only carry her guns into the fight but she must nurse them through all its savage hammering, so effectually that they shall be able to pour shell into the enemy until they have silenced or sent him to the bottom.



THE ARMOR PLAN OF THE "NEVADA" AND "OKLAHOMA," THE TWO MOST POWERFULLY PROTECTED SHIPS YET DESIGNED.

So let us see how these conditions have been met in our new ships. Taking the *North Dakota*, for instance, as a basis of comparison, we find that the armor protection has been entirely removed from the secondary battery of 5-inch guns—a wise step, which might well have been taken several years ago. For it is a fact that the 5, 6 or 7 inches of armor with which the secondary batteries of warships of today are protected, will simply serve as a shell burster, delaying the high-explosive 14-inch shells long enough to cause the little firing hammer within the shells to leap forward and detonate the high explosive, the burst taking place after the shell has passed through the armor and is well within the body of the ship. So the torpedo-defense guns will have nothing in front of them except the ordinary ½-inch or ⅝-inch plating of the ship's side, which may very well allow the shells to pass through without bursting among the gun crews crowded about the guns.

The most important armor on a ship is undoubtedly the belt armor upon the hull itself; for to this is committed the duty of keeping the ship afloat and preventing projectiles from striking a vital blow in the magazine, boiler rooms or engine rooms. In the new ships the belt will be 17½ feet in width and, at mean draught, it will extend from 9 feet above to 8 feet 6 inches below the water. It will have the unprecedented thickness of 13½ inches, which it will maintain from its upper edge down to within a few feet of its bottom, where it will begin to taper to a minimum width at the bottom of 8 inches. Very rarely, if ever, will the bottom edge of this deep belt be rolled out of water, exposing the thin plating below. This belt will extend for over 400 feet along each side of the ship. It will terminate well forward of No. 1 barbette, where it will be carried, with the same depth and thickness, entirely across the ship. At its after end the belt armor will be carried at its full depth of 17½ feet to a point about 30 feet aft of No. 4 barbette. Here there will be a jog, the depth of the belt decreasing from 17½ feet to 8½ feet, at which depth it will be continued aft for another 60 feet. Transverse bulkheads of the same thickness as the belt will here be carried across the ship.

An important feature of the side armor is the manner in which the plating will be laid on the ship. Hitherto the armor has been placed horizontally, in two strips, with a continuous horizontal joint, located slightly above the water line, between the upper and lower strip. This had the disadvantage that it presented a continuous line of cleavage, near the water line, and, therefore, at a most vulnerable point. In the new ships the armor plates are laid vertically, the joints being vertical and the plating extending the whole depth of the belt without any continuous joint at the water line. This is a most important improvement which will add greatly to the protective power of the side armor. Associated with the heavy belt in the work of protecting the ship's stability are two protective decks—a lower deck 1½ inches thick on the flat which will slope along the sides to a junction with the bottom of the armor plate 8½ feet below the water line. The slopes of this deck are 2 inches in thickness. On the deck above (the gun deck) is an upper protective deck, 3 inches in thickness. These two decks provide an excellent protection against plunging fire, and also against fragments of shells which might be exploded in passing through the thin ship's plating in the wake of the gun deck.

Equally massive is the armor protection for the main gun positions. The barbette armor extends, with a thickness of 13 inches, from the turret down to the upper protective deck, and from the upper to the lower protective deck the thickness is reduced to 4½ inches—this because of the 13-inch protection afforded by the side armor. The turret armor is equally massive. The port plate is 16 inches on the two-gun turrets and 18 inches on the three-gun turrets, and the side and rear armor is 10 and 9 inches in thickness, while the roof carries 5 inches of armor.

The battle of the Sea of Japan showed how important it is to thoroughly protect the positions from which the fighting of the ship is controlled, and particular attention has been given to this in our new design. The conning tower and the signal station back of it each carry no less than 16 inches of armor, and to protect the communications—telegraph and telephone wires, voice tubes, etc.—the section upon which conning tower and signal station are supported has walls of 16-inch armor, which are carried down to the protective decks.

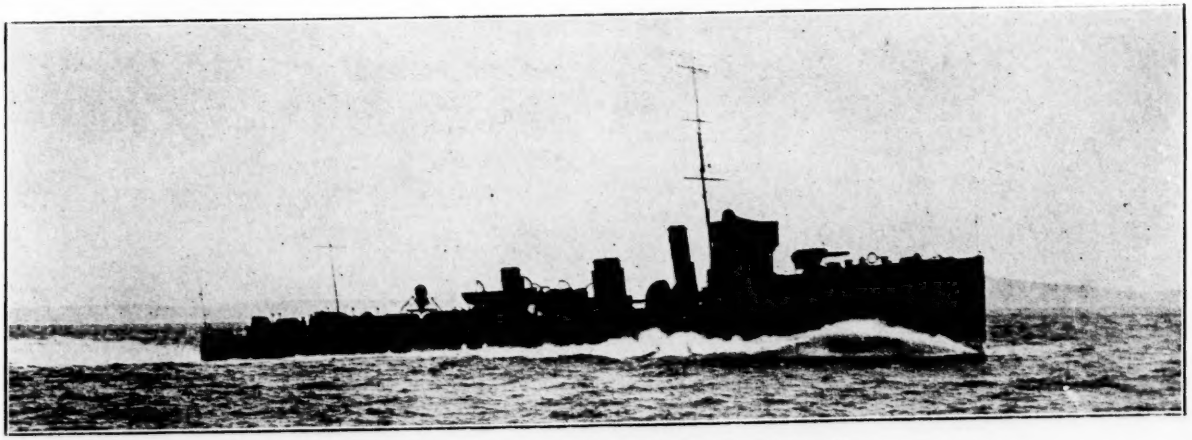
It will be noticed that the new ships have but one smokestack—and thereby hangs a tale. The new ships, as already stated, will burn fuel oil exclusively. This has enabled the designer to dispense entirely with coal bunkers—the oil being carried chiefly in the double bottom of the ship. The omission of bunkers sets free a large amount of space below decks, which has enabled the designer to concentrate all of the six boiler compartments at the center of the ship, where they occupy only 65 feet of her length. Hence, it was possible to use a single smokestack, placed immediately above the boiler rooms, and hence, again—and this is the important point—it was found possible to place around the whole of the uptakes a massive redoubt of inclined armor with walls everywhere 13 inches in thickness. This redoubt extends from the upper protective deck to the spar deck, and that portion of the smokestack and uptakes which is within the structure of the ship will be completely protected against perforation. The importance of this construction will be appreciated, when we bear in mind that, in the Japanese war, it was the perforation of the uptakes which contributed largely to the collapse of the Russian ships. The poisonous gases, escaping between decks, were drawn down and disseminated throughout the ship, frequently driving the crew from their quarters.

From the above description it will be evident that in the *Nevada* and *Oklahoma* the United States Navy will possess two fighting ships which will be the equal, if not superior, to any ships in their gun power and which will be greatly superior in their power of endurance in a long-drawn-out fight. If Congress will only be wise enough to add year by year the two battleships which represent the minimum requirement of our Navy, we shall be in a position to maintain our standing among the navies of the world. If less than two battleships a year be authorized, our Navy will steadily retrograde.—"Scientific American."

ENGLAND.

H. M. TORPEDO-BOAT DESTROYER *ARCHER*.

The *Archer* has a length of 240 feet, a beam of 25 feet 7 inches, and a total weight of hull of 325 tons. With a draught of 8 feet the displacement tonnage is about 785 tons. She is fitted with Yarrow boilers with superheaters in the uptakes. On the full-speed trial of the *Archer* in January, 1912, the degree of superheat of the turbines was 94 degrees F. The turbines developed slightly over 18,500 shaft horsepower, and the mean speed upon the eight hours' continuous trials was 30.3 knots. Oil alone is used in the boilers, and at full speed the consumption was at the rate of 1 pound per shaft horsepower per hour, while at the cruising speed of 13 knots the consumption was 17.4 tons per 24 hours, giving a radius of action of 3,040 nautical miles—a very satisfactory result. The *Archer* is built to maintain a high speed against heavy seas, having a high forecastle. She is fitted with two 4-inch breech-loading guns and two 12-pounder 12-cwt. quick-firing guns.

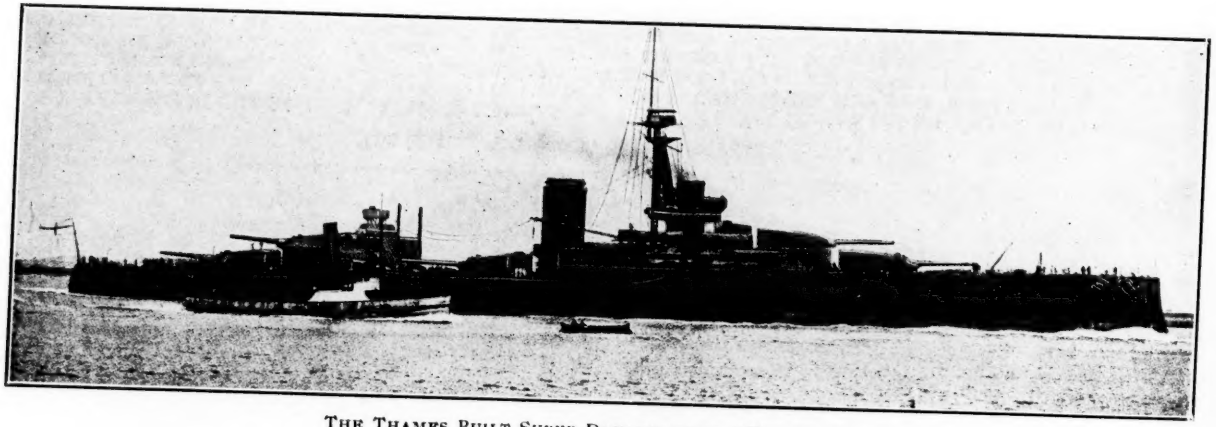


HIS MAJESTY'S TORPEDO-BOAT DESTROYER "FURY."

The destroyer illustrated is the *Fury*, built by Messrs. A. and J. Inglis, Limited, Glasgow, and is typical of twenty vessels constructed at the same time by various builders from designs by Sir Philip Watts, K.C.B., Director of Naval Construction.

These vessels were specially designed for maintaining a high speed in a heavy seaway, and were thus constructed with a high forecastle and bridge, one of the two 4-inch guns, with which they are armed, being placed on this forecastle, while the other is mounted aft. In addition there are two 12-pounder 12-cwt. quick-firing guns amidship, with two torpedo-launching tubes. The length of the hull is 240 feet, the breadth 25 feet 6 inches, and the depth 15 feet 7 inches, the weight of hull being about 310 tons. On a draught of 7 feet 10 inches the trial displacement was 770 tons.

The boilers are of the Yarrow type, fired by oil fuel, and the view given, showing the vessel steaming at full speed, indicates the success which has been achieved in respect of absence of smoke. The machinery, of the Parsons turbine type, was designed to develop 13,500 shaft horsepower, from which the speed expected was 27 knots. On the full-power trial, however, the shaft horsepower was about 14,000, and the mean speed during the eight hours was 27.82 knots. For one hour the vessel ran at 28.2 knots. On the fuel-consumption trial at 13¼ knots with 1,000 horsepower it was found that the radius of action was 2,600 nautical miles.



THE THAMES BUILT SUPER-DREADNOUGHT "THUNDERER."

STEAM TRIALS OF H.M.S. *THUNDERER.*

The battleship *Thunderer* has successfully passed all her steam trials, and will return to the Thames to be completed for commission. This vessel was built by the Thames Iron Works and Shipbuilding Company, the first keel plate having been laid on April 13, 1910, and the vessel launched on February 1, 1911. The date for completion is May next. The *Thunderer* is a battleship with a displacement tonnage of 22,500 tons, the length being 545 feet, and the beam 88½ feet. She is armed with ten 13.5-inch and twenty-four 4-inch guns, the gun positions being protected by 10-inch armor. The main broadside belt is 12 inches thick, tapering forward and aft to 4 inches, while the upper part on the broadside is protected with 9-inch armor. The boilers are of the Babcock & Wilcox type, and the turbines of the Parsons design.

The first official trial consisted of a thirty hours' run at about 70 per cent. of the full designed power. During part of the time, as is usual with such ships, the engines worked with open and closed exhaust, separate results being taken in connection with the design of machinery for future ships. The mean result of the thirty hours' run was that the machinery developed 18,927 shaft horsepower, while the average revolutions were 265 per minute. Six runs were made on the measured mile at Polperro, when, with a mean of 17,787 shaft horsepower, a speed of 18.8 knots was realized.

Throughout the eight hours' trial at full power the engines ran at an average of 299 revolutions per minute, the mean power developed being 27,416 shaft horsepower, while the coal consumption was 1.78 pounds per shaft horsepower per hour. On this run also the ship made six runs over the measured mile, the mean results being: Revolutions, 300; shaft horsepower, 27,426; and speed, 20.8 knots. The machinery throughout worked satisfactorily.—"Engineering."

FRANCE.

TRIALS OF THE FRENCH ARMORED CRUISER *WALDECK ROUSSEAU.*

This vessel is 515 feet 10 inches long, and has a beam of 70 feet 7 inches and a draught of 27 feet 4 inches. Her displacement is 14,158 tons. She is fitted with three sets of triple-expansion engines having a high-pressure cylinder 46½ inches in diameter, an intermediate-pressure cylinder of 68 inches, and two low-pressure cylinders of 78 inches, the stroke in all cases being 41 1/3 inches. There are forty-two Niclausse water-tube boilers. The combined grate area is 2,669 square feet, and the total heating surface 94,473 square feet. The normal working pressure is 17 kilogrammes per square centimeter—say, 242 pounds per square inch.

10 *Hours' Full-Speed Trial.*

Indicated horsepower, mean	36,110
Average speed, knots	23.10
Number of revolutions, mean per minute	129.2
Consumption of coal per square foot of grate area, pounds	28.47
Consumption of coal per I.H.P. per hour, pounds	1.94

3 *Hours' Full-Speed Trial with Three-quarters of the Boilers.*

Indicated horsepower, mean	36,000

Average speed, knots	23.15
Number of revolutions, mean per minute	129
Consumption of coal per square foot of grate area, pounds	39.73
Consumption of coal per I.H.P. per hour, pounds, nearly	3

24 Hours' Trial with Normal Draft.

Indicated horsepower, mean	28,300
Average speed, knots	21.62
Consumption of coal per square foot of grate area, pounds	15.52
Consumption of coal per I.H.P. per hour, pounds	1.5

Cruising Consumption Trial.

Indicated horsepower, mean	14,038
Speed, knots	18.48
Coal consumption per I.H.P. per hour, pounds	1.5

OBITUARY.

CAPTAIN THOMAS FREDERICK CARTER.

By Walter M. McFarland, Member.

In order that a highly complex organization like the Navy may maintain the splendid reputation for efficiency which it has so thoroughly earned, it is requisite that the individual officers should all be men of fine character and high professional attainments, and a necessary consequence is that the great majority can not achieve special distinction, but must be content with the knowledge of duty always well done and the esteem of their brother officers. What would constitute eminence in a less distinguished company here becomes simply the warrant for a place on the roll of officers who have always maintained the highest traditions of the Service. Such a man was my dear old friend and classmate, Captain Carter, and I count it a privilege to pay my tribute to his noble character and devotion to duty.

He was a member of one of the early classes of Cadet-Engineers and had the benefit of training under those pioneers of engineering education, John C. Kafer, Chas. H. Manning, David P. Jones, Wm. L. Nicoll, Robert Crawford and others. Probably none of us appreciated them at their real worth when we were their pupils, but, as the years have passed and some of us have taken our turn as instructors in engineering, we have learned to know what splendid work they did, almost without text books and entirely without precedents in technical education. The high reputation of these pupils in the engineering world is the best monument to their efficiency as teachers.

Captain Carter's whole life was spent in engineering work, ashore and afloat, some of it routine and some very interest-

ing, including a tour as instructor at the Naval Academy. Possibly the most interesting was his work as inspector of machinery for the *Brooklyn,* followed by service on that vessel. This included a trip to Europe at the sixtieth anniversary of the accession of Queen Victoria in 1897, when the *Brooklyn* attracted great attention; followed the next year by service in the War with Spain and the Battle of Santiago. Here, as always, his duty was performed with the highest efficiency. Other shore duty covered several tours as inspector of material and one as Chief Engineer of the Pensacola Navy Yard.

During his last tour of duty, as inspector of material in the Pittsburgh district, we were much together and often discussed his work, so that I had a chance to appreciate his thorough grasp of the subject, the splendid judgment shown in handling delicate situations which arose and his conscientious discharge of duty. My acquaintance among the manufacturers whose product he inspected was wide, and they all bore testimony to his fairness and helpfulness to them while absolutely conscientious in maintaining the interests of the Government. "He is the best inspector I have ever known," was a common remark. I know of cases where his thorough knowledge of manufacture enabled him to point out the cause of troubles, so that they might be removed, the product brought to a high degree of excellence, and the loss due to rejections avoided.

He was an ideal friend, loyal and affectionate, and counting no service too great a demand upon him. His personality was very attractive and all who met him soon discovered his fine and noble character and were glad to be counted among his friends; those in the Service will, I am sure, feel that this tribute does him scant justice.

It is pleasant to record that his own beautiful character found a congenial atmosphere in his home life, where his wife and daughter were as charming in their way as he in his.

It is possible that Captain Carter was a victim of the Spartan system of exercise which obtained a few years ago. Preparation for and performance of the fifty-mile walk seems to

have damaged his heart, and this led to his retirement in 1910. He continued on duty for a time, but had been failing for some months prior to his death, April 4, 1912. Letters from him received during this period were always cheerful, with little about himself and much of kindness about others.

Thus has passed away a competent and efficient officer, a steadfast friend, and a man of noble and unselfish character, who made those about him happier for having known him, and whose memory will long be cherished as an exemplar of all that is best in the naval officer.

ASSOCIATION NOTES.

The banquet of the Society was held at Rauscher's, Washington, D. C., on March 9, 1912, the fiftieth anniversary of the battle of the *Monitor* and *Merrimac*. About 200 members and guests were in attendance. The toasts were as follows:

"Naval Engineering from the *Monitor* to the *Oklahoma*," Mr. Walter M. McFarland.

"A Powerful Navy at Once the Guarantee and Exponent of a Nation's Prosperity," Hon. W. C. Redfield, M. C.

"The Navy and Congress," Hon. Albert Estopinal, M. C.

"The Importance of Naval Engineering," Hon. George v. L. Meyer, Secretary of the Navy.

"The Navy and the People," Hon. Joseph Cannon, M. C.

The Toastmaster was Captain William Strother Smith, U. S. N.

The committee in charge of the banquet was as follows:

Captain William Strother Smith, U. S. Navy; Engineer-in-Chief C. A. McAllister, U. S. R. C. S.; Lieutenant N. H. Wright, U. S. Navy.

The following new members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Lauriat, Philip W., First Lieutenant, U. S. R. C. S.
Roemer, Charles G., Second Lieutenant, U. S. R. C. S.
Wishaar, William P., Second Lieutenant, U. S. R. C. S.

ASSOCIATES.

Drake, George B., N. A., 17 Battery Place, New York City.
Gatewood, William, 327 Fifty-second Street, Newport News, Va.

Gauntlett, Fred J., The Parkwood, K Street, N. W., Washington, D. C.

Hamilton, Brace Hayden, Westinghouse Electric and Manufacturing Company, Hibbs Building, Washington, D. C.

Hekking, M. J., Office of Inspector of Machinery, New York Shipbuilding Co., Camden, N. J.

Hutchison, M. R., Chief Engineer to and Personal Representative of Thomas A. Edison, Orange, N. J.

Thayer, Horace H., Jr., Engineer and N. A., 1012 King Street, Wilmington, Del.

White, William Albert, Resident Manager, Washington Engine Works, Washington and Leroy Streets, New York City.

Gauntlett, Fred J., The Parkwood, K Street, N. W., Washington, D. C.

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Reimer, Charles O., Second Lieutenant, U. S. R. C. S.
Webber, William T., Second Lieutenant, U. S. R. C. S.

ASSOCIATES.

Brake, George R., N. A., 17 Battery Place, New York City.
Galewood, William, 167 Fifty-second Street, Newport News, Va.